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AMRL-TDR-62-89

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EXPLORATORY INVESTIGATION OF THE MAN AMPLIFIER CONCEPT

TECHNICAL DOCUMENTARY REPORT NO. AMRL-TDR-62-89

August 1962

Behavioral Sciences Laboratory
6570th Aerospace Medical Research Laboratories
Aerospace Medical Division
Air Force Systems Command
Wright-Patterson Air Force Base, Ohio

Contract Monitor: Leroy D. Pigg, Major, USAF
Project No. 7184, Task No. 718406

[Prepared under Contract No. AF 18(600)-1922
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20030113012

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FOREWORD

This research was conducted by Cornell Aeronautical Laboratory, Inc., under Contract AF 18(600)-1922 in support of Project No. 7184, "Human Performance in Advanced Systems," Task No. 718406, "Design Criteria for Ease of Maintenance." This contract was supported by the Bioastronautics Office of the Air Force Systems Command, Andrews Air Force Base, Washington, D. C., and monitored by the Behavioral Sciences Laboratory, 6570th Aerospace Medical Research Laboratories, Aerospace Medical Division, Air Force Systems Command. The research was conducted between July 1961 and April 1962. The contract monitor was Major Leroy D. Pigg, Maintenance Design Section, Human Engineering Branch, Behavioral Sciences Laboratory.

The authors gratefully acknowledge the cooperation and aid given them by personnel of the Anthropology Section, Human Engineering Branch, Behavioral Sciences Laboratory, who conducted tests and furnished the experimental data on human-joint angular velocities contained in this report.

This is the final report under this contract and in accordance with Cornell Aeronautical Laboratory, Inc., procedures it is cataloged as Report No. VO-1616-V-1

ABSTRACT

Preliminary investigations were conducted to ascertain and define some of the major problems and uncertainties requiring additional research before feasibility of the Man Amplifier concept can be evaluated. Study areas included possible Air Force applications as a basis for selecting a maximum load-carrying capability, human factors from the standpoints of body kinematics and physical anthropology, structures and mechanical design, and servo system and power requirements. The dynamic response characteristics of an elbow-joint amplifier, as determined theoretically and experimentally, were compared. Comparison of position tracking tests performed both with and without power boost provided by the elbow-joint servo indicated that employment of power boost did not increase the tracking error above that exhibited by the unaided operator. It was concluded that: (1) duplication, in the Man Amplifier, of all the human joint motion capability is impractical; (2) experimentation is necessary to determine the essential joints, motion ranges, and dynamic responses; (3) the inability to counter the overturning moments will, in many instances, limit the load-handling capability; (4) conventional valve-controlled hydraulic servos are unsuitable for the Man Amplifier, (5) particularly difficult problems will be encountered in the general areas of mechanical design, sensors, and servomechanisms. Some specific tasks that should be undertaken to assess engineering feasibility of the concept are outlined.

PUBLICATION REVIEW

This technical documentary report has been reviewed and is approved.

Walter F. Grether

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Technical Director
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INTRODUCTION

Modern man, with his reasoning power, memory and nervous system, is capable of controlling forces and machines of tremendous size and speed. On the other hand, in many instances, his muscular development, from the standpoints of strength and endurance, does not adequately complement his highly refined intelligence. The Man Amplifier principle conceivably would alleviate this mismatch of human intelligence and muscular power.

The Man Amplifier, conceived by Cornell Aeronautical Laboratory, Inc. (CAL), is an exoskeleton employing powered joints that is worn by a man to augment or amplify his muscular strength and to increase his endurance in the performance of tasks requiring large amounts of physical exertion.

An artist's interpretation of the Man Amplifier applied to military use is depicted in figure 1. It consists of a basic structural exoskeleton with appropriate articulated joints compatible with those of man. All external loads as well as the weight of the Man Amplifier itself are borne by this structural skeleton. Each joint is powered by one or more servomotors which provide the necessary torques and power boost for the device. These servomotors respond to the outputs of suitable sensors linking the man to the machine and cause the links of the mechanism to follow the natural motions of their human counterparts with little or no conscious effort on the part of the operator. A portable, self-contained power pack attached to the back of the exoskeleton provides the power necessary to operate the device.

Results of preliminary investigations of the Man Amplifier concept are presented in this report. These investigations consisted of: (1) preliminary analytical studies to define some of the major problem areas which must be examined in detail before the technical feasibility of the concept can be established, and (2) experiments, using the CAL elbow-joint amplifier, to obtain a preliminary indication of man-machine compatibility. Technical areas investigated include: (1) applications of the device in Air Force operations, (2) human engineering as characterized by physical anthropology and kinematics of the human body members, (3) structures and mechanical design, and (4) servomechanism and joint power requirements.

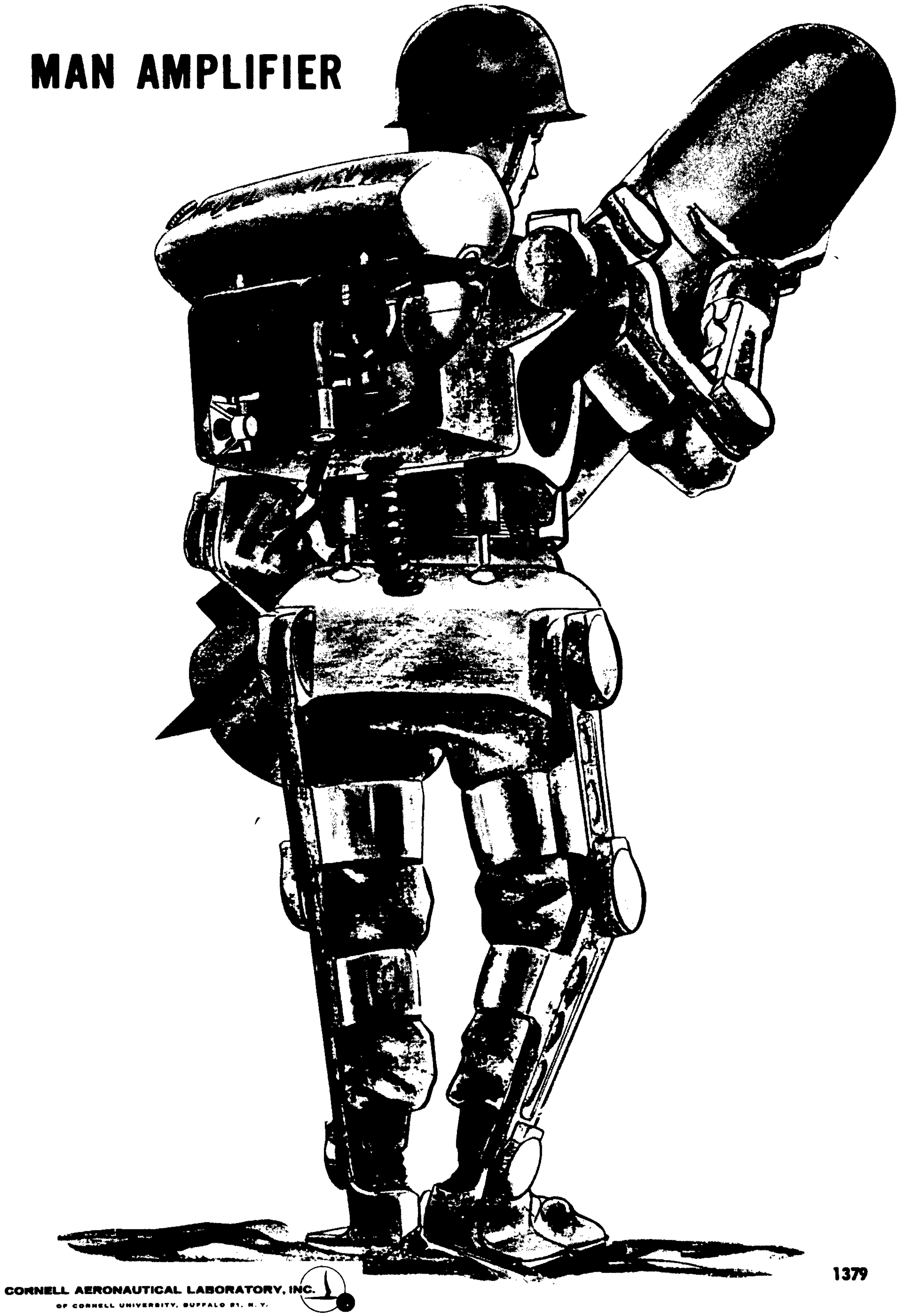


Figure 1 ARTIST'S CONCEPTION OF MAN AMPLIFIER

TECHNICAL STUDIES

Man Amplifier Applications in Air Force Operations

At the present time, no branch of the military services has defined and/or issued a formal requirement for the Man Amplifier system. Likewise, this study program was not intended to include operational analyses from which justifiable reasons for developing the concept might be set forth. However, it was appropriate to consider typical tasks, i. e., what manner the Man Amplifier could be utilized to advantage by the Air Force in its various operations, to aid in the definition of preliminary requirements for the Man Amplifier, particularly with regard to its load carrying capacity.

Most tasks presently envisioned for the Man Amplifier involve lifting and/or moving objects which are too heavy for the unaided man. It was logical, therefore, to examine some of the various types of mechanical equipment included in the Air Force inventory for handling and transporting material to determine those whose functions might be performed by a Man Amplifier.

Descriptions of at least a part of the Air Force inventory of cargo handling equipment such as forklift trucks, dollies, cranes, trailers, jacks, mechanized pry bars, etc. may be found in reference 1. It appears that, from the standpoint of maximum load capacity, only a few of these pieces of equipment could be replaced by an "amplified" man. For example, forklifts and warehouse crane trucks have capacities ranging from 4000 lbs. to 10,000 lbs. or more. These capacities are much greater than those that thus far have been considered for a single Man Amplifier.

The equipment used in present cargo handling systems is frequently heavy, bulky, and generally limited to operation on firm, prepared surfaces. At well-established air bases located, for example, in the zone of interior, these large capacity vehicles and equipments are needed and may be found in quantity. For these bases there is little difficulty in supplying cargo-handling and transporting equipment in sufficient quantity and variety to carry out the necessary base operations. Furthermore, these installations, having paved roads and large, easily accessible storage and work areas, are well-suited to the use of these vehicles.

However, one can readily appreciate the value of the Man Amplifier at a remote, perhaps temporary, base of operations where the logistics problem becomes more acute. Large quantities of supplies of all types are required that must be airlifted to the remote base and compete for air transport. It is clear that the luxury of an abundance of special-purpose equipment to perform the many diversified tasks at the remote base can be ill-afforded and the versatility and adaptability of the Man Amplifier would be advantageous for the performance of those tasks not requiring the maximum load capacity of conventional cargo-handling vehicles. These tasks may include such things as clearing away small trees and obstacles, erection of shelters, positioning of equipment and supplies (communications, field kitchens, fuel drums, etc.), laying of steel-mesh portable runways, loading armament and munitions on aircraft, and emergency rescue. The capability of threading through forests and over rough terrain while carrying heavy loads would be advantageous as, for example, in the retrieval of air-dropped supplies that land in places inaccessible to conventional vehicles.

The tasks cited above do not establish a firm requirement for the load-carrying capacity of the Man Amplifier; therefore the maximum value is, within reasonable limits, somewhat arbitrary. For these preliminary studies, a 1500 lb. load was selected as a maximum. As will be discussed in a subsequent section, this value is probably somewhat greater than the maximum weight that the Man Amplifier would be capable of handling in the performance of a typical lifting task. However, a 1500 lb. capacity would enable two Man Amplifiers, working as a team, to match the 3000 lb. capacity of the MA-1 bomb-lift truck. This vehicle is used by the Air Force to lift, carry, and load externally-carried munitions, weapons, jato bottles, rockets, fuel tanks, etc. on aircraft. It also may be used as a general cargo-handling and utility vehicle when fitted with special attachments.

Human Body Kinematic and Anthropometric Considerations

The unique feature of the Man Amplifier in comparison with other man-machine systems is that the mechanism is installed on and worn by the human operator and hence must follow his principal body motions. It is obvious therefore, that much information about the human body is required before an exoskeleton can be designed to fit around and move with the human frame.

An important factor to be considered is variability of body dimensions of individuals using the device. It will not be satisfactory to size the various links and joints comprising the exoskeleton according to the dimensions of an "average" man because, as is so pointedly shown in reference 2, no such creature exists. It is also impractical, from an economic viewpoint, to custom fit each machine according to the size requirements of each individual operator. Clearly then, some means for adjusting the Man Amplifier skeletal structure to accommodate a range of human body dimensions is required.

Basically, there are two factors which contribute to the sizing problem. These may be thought of as the adjustability needed to account for differences in body build (e.g. rotund, thin, muscular) and that necessary as a result of differences in skeletal frame dimensions (e.g. link lengths or distances between joint centers). To illustrate the dimensional variability found in adult human males, some selected anthropometric data are presented in tables I and II (references 3, 4).

In addition to the linear dimensions of the various body members, consideration must be given to the angular motion ranges permitted by the various joints. A study was made to obtain information on the motion ranges of human joints to help in the establishment of preliminary estimates of the joint-range requirements of the Man Amplifier skeletal structure.

TABLE I

TYPICAL ANTHROPOMETRIC DATA SHOWING VARIABILITY OF BODY DIMENSIONS *

Measurement	Range	Mean	Standard Deviation	Percentiles		
				5th	50th	95th
Shoulder (acromial) Height	47.24-64.17	56.50	2.28	52.8	56.6	60.2
Waist Height	34.65-48.82	42.02	1.81	39.1	42.1	45.0
Kneecap (patella) Height	15.75-23.23	20.22	1.03	18.4	20.2	21.9
Shoulder-Elbow Length	11.42-18.11	14.32	0.69	13.2	14.3	15.4
Forearm-Hand Length	15.35-22.05	18.86	0.81	17.6	18.9	20.2
Shoulder Breadth	14.57-22.83	17.88	0.91	16.5	17.9	19.4
Chest Breadth	9.45-15.35	12.03	0.80	10.8	12.0	13.4
Hip Breadth	8.27-15.75	13.17	0.73	12.1	13.2	14.4
Chest Depth	6.69-12.99	9.06	0.75	8.0	9.0	10.4
Buttock Depth	6.30-11.81	8.81	0.82	7.6	8.8	10.2
Chest Circumference	31.10-49.61	38.80	2.45	35.1	38.7	43.2
Buttock Circumference	29.92-46.85	37.78	2.29	34.3	37.7	41.8
Thigh Circumference	14.57-28.74	22.39	1.74	19.6	22.4	25.3
Calf Circumference	9.84-18.50	14.40	0.96	12.9	14.4	16.0
Biceps Circumference (flexed)	8.27-16.93	12.79	1.07	11.2	12.8	14.6
Lower Arm Circumference (flexed)	8.66-15.35	11.50	0.73	10.4	11.5	12.7

NOTE: All dimensions in inches

* Data From WADC TR 52-321 (ref. 3)

TABLE II

ESTIMATION OF LINK DIMENSIONS OF AIR FORCE FLYING PERSONNEL BASED ON RATIOS FROM CADAVER MEASUREMENTS*

LINK	LENGTH (cm)		
	Percentile		
	5th	50th	95th
Clavicle (sternoclavicular to claviscapular joint centers)	13.1	14.1	15.2
Scapula (claviscapular to glenohumeral joint centers)		3.5	
Humerus (glenohumeral to elbow joint centers)	28.6	30.2	32.0
Radius (elbow to wrist joint centers)	25.7	27.2	28.5
Transpelvic		17.1	
Femur (hip to knee joint centers)	40.5	43.4	46.0
Tibial (knee to ankle joint centers)	38.0	40.9	43.9
Vertical distance from midtalus to floor level		8.2	

* Values from WADC TR 55-159 (ref. 4)

The determination of the motion ranges of human joints has been the objective of many investigators. Results obtained by these investigators are not always comparable, however, due to differences in (1) subject characteristics (live subjects or cadavers, age, physique, etc.), (2) subject position and type of motion, and (3) methods and techniques used in obtaining the measurements.

The mean values of joint motion ranges given in the following paragraphs were obtained from references 5 and 6. Data cited from the former are "considered average for each joint as stated but may vary slightly from other published figures." Data obtained from reference 6 are based on actual measurements on a sample of 39 young men. Values for the standard deviation are those given in reference 6 but may only be considered to be approximate with respect to the joint range data of reference 5.

Pictorial representations of the types of joint motion from the neutral positions are shown for clarification. In addition, medical terminology used in the description are translated into "layman's language" to facilitate understanding by those unfamiliar with terms of that particular discipline.

1. Spine (fig. 2a)

Motions of the spine are a result of the sum of motions which take place at the articulation between each of the vertebrae.

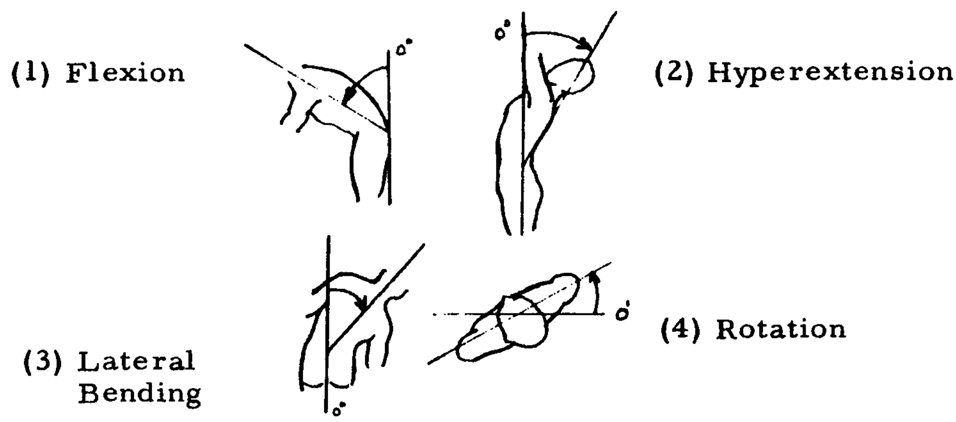


Figure 2a Motions at Spine

Flexion (1) (bending forward)	Mean 70° (ref. 5)
Hyperextension (2) (bending backward)	Mean 30° (ref. 5)
Lateral bending (3) (each side of neutral)	Mean 40° (ref. 5)
Rotation (4) (each side of neutral)	Mean 35° (ref. 5)

2. <u>Shoulder</u> (fig. 2b)

Motion at the shoulder is a result of movement at several joints in the shoulder area. It is described in reference 5 as, "Movements at the shoulder joint take place between the head of the humerus (upper arm bone) and the glenoid cavity of the scapula (shoulder blade) together with scapulathoracic (shoulder blade-chest), acromioclavicular (shoulder blade-collar bone) and sternoclavicular (breast bone-collar bone) motion."

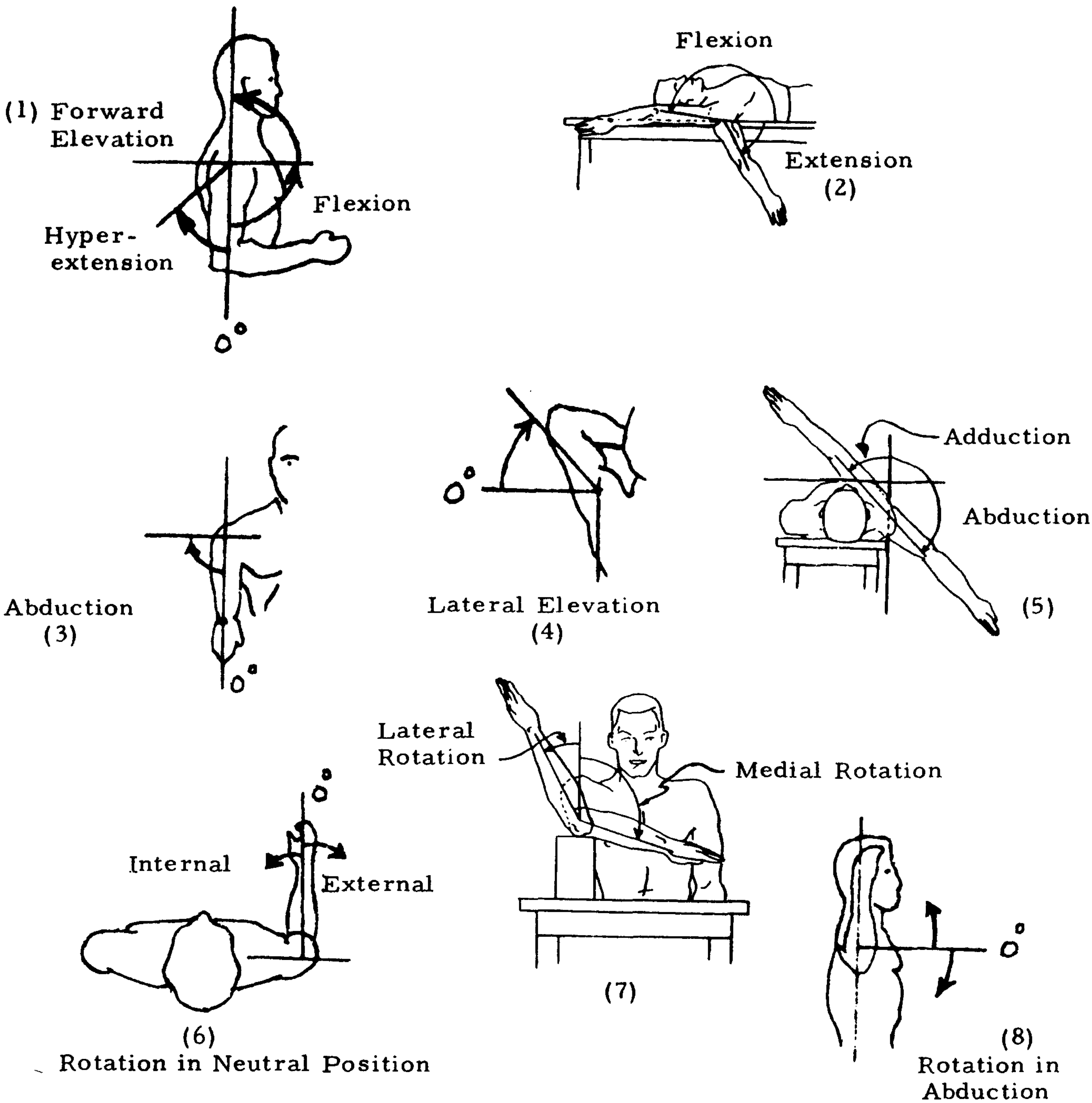


Figure 2b Motions at Shoulder

Flexion (1) (upper arm moves forward and upward in sagittal plane)	Mean 90° (ref. 5)
Forward elevation (1) (continuation of flexion motion to limit)	Mean 90° (ref. 5)
Flexion (2) (arm moves upward and backward in sagittal plane)	Mean 188° S. D. 12° (ref. 6)
Hyperextension (1) (arm moves backward in sagittal plane)	Mean 45° (ref. 5)
Extension (2) (arm moves downward and backward in sagittal plane)	Mean 61° S. D. 14° (ref. 6)
Abduction (3) (upper arm moves outward and upward from the side)	Mean 90° (ref. 5)
Lateral elevation (4) (continuation of upward motion of arm from fully abducted position)	Mean 40° (ref. 5)
Abduction (5) (arm moved laterally to limit of abduction and perpendicular to long axis of body)	Mean 134° S. D. 17° (ref. 6)
Adduction (5) (arm swung maximally across the chest to the opposite side of the body)	Mean 48° S. D. 9° (ref. 6)
Internal rotation in neutral position (6) (forearm swung in horizontal plane inward across the body)	Mean 90° (ref. 5)
External rotation in neutral position (6) (forearm swung in horizontal plane outward from the body)	Mean 45° (ref. 5)
Medial rotation (7) (upper arm flexed 90°, forearm vertical and swung toward midline)	Mean 97° S. D. 22° (ref. 6)
Lateral rotation (7) (same as above except forearm swung outward from midline)	Mean 34° S. D. 13° (ref. 6)
Internal rotation in abducted position (8) (forearm moved downward from 90° abduction position)	Mean 90° (ref. 5)
External rotation in abducted position (8) (forearm moved upward from 90° abduction position)	Mean 90° (ref. 5)

3. <u>Elbow</u> (fig. 2c)

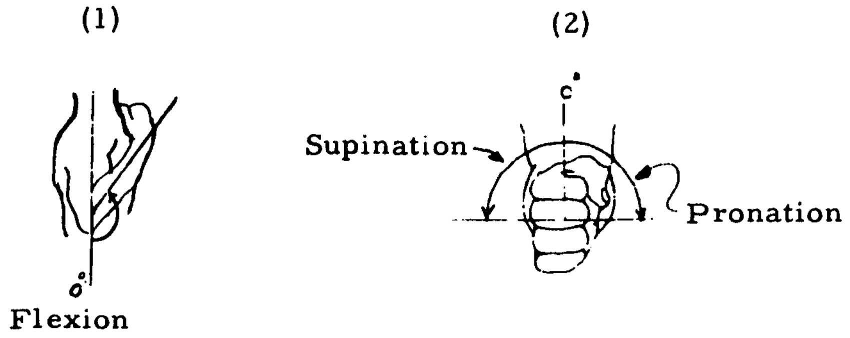


Figure 2c Motions of Elbow

Flexion (1) (bending of forearm on upper arm)	Mean 145°	(ref. 5)
	Mean 142° S. D. 10°	(ref. 6)
Supination (2) (forearm placed with ulnar border of hand, little finger side down, and rotated to palm up)	Mean 90°	(ref. 5)
	Mean 113° S. D. 22°	(ref. 6)
Pronation (2) (same as above only hand rotated palm down)	Mean 90°	(ref. 5)
	Mean 77° S. D. 24°	(ref. 6)

4. <u>Wrist</u> (fig. 2d)

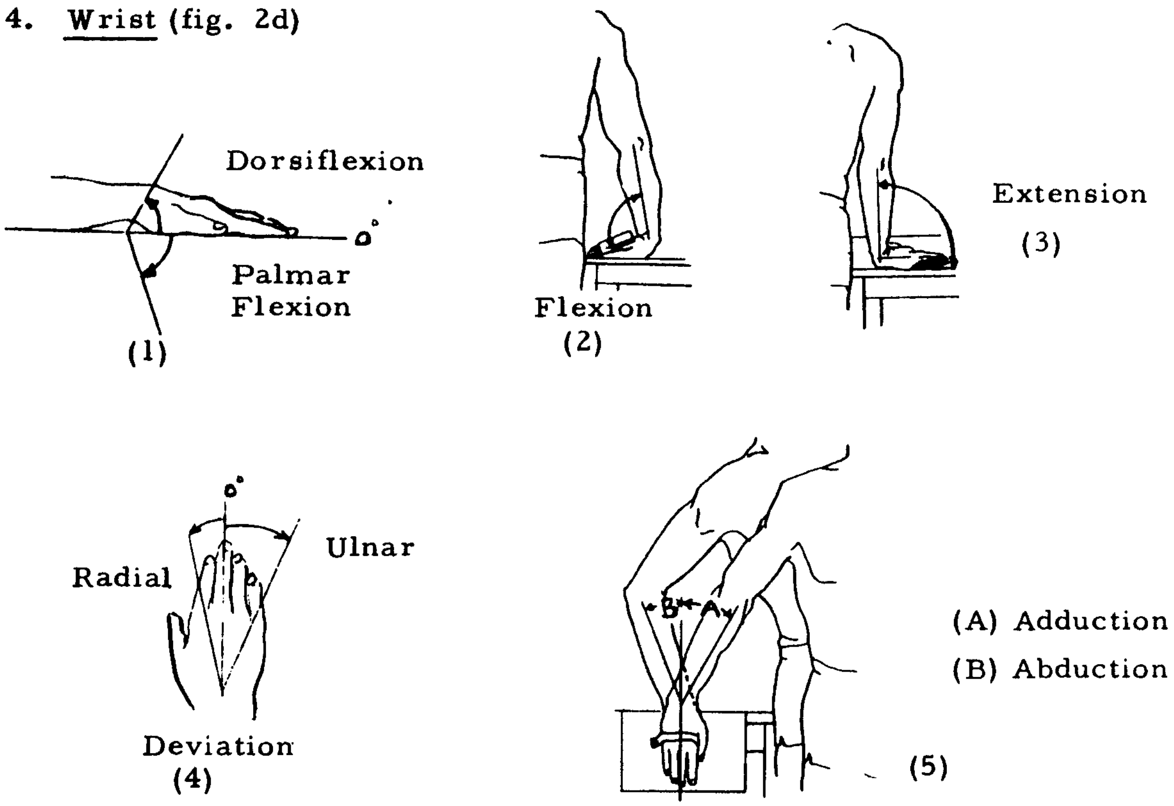


Figure 2d Motions of Wrist

Palmar Flexion (1) (extended hand, in line with forearm, is bent at wrist palm downward)	Mean 70° (ref. 5)
Dorsiflexion (1) (same as above except hand is bent up)	Mean 65° (ref. 5)
Flexion (2) (dorsal surface of hand on table, forearm supinated and maximally flexed at wrist)	Mean 90° S. D. 12° (ref. 6)
Extension (3) (with palm down on table, fingers pointing away from body, supinated forearm is maximally bent at wrist)	Mean 99° S. D. 13° (ref. 6)
Ulnar deviation (4) (extended hand, in line with forearm, is bent sidewise outward)	Mean 30° (ref. 5)
Radial deviation (4) (same as above except hand is bent sidewise inward toward midline)	Mean 15° (ref. 5)
Adduction (5) (hand with fingers pointing down held flat and vertical by restraining gear, forearm bent at wrist towards the body)	Mean 47° S. D. 7° (ref. 6)
Abduction (5) (same as above except forearm bent at wrist away from the body)	Mean 27° S. D. 9° (ref. 6)

Some pronation-supination motion occurs at the wrist but has been included in the values given for the elbow.

5. Hip (fig. 2e)

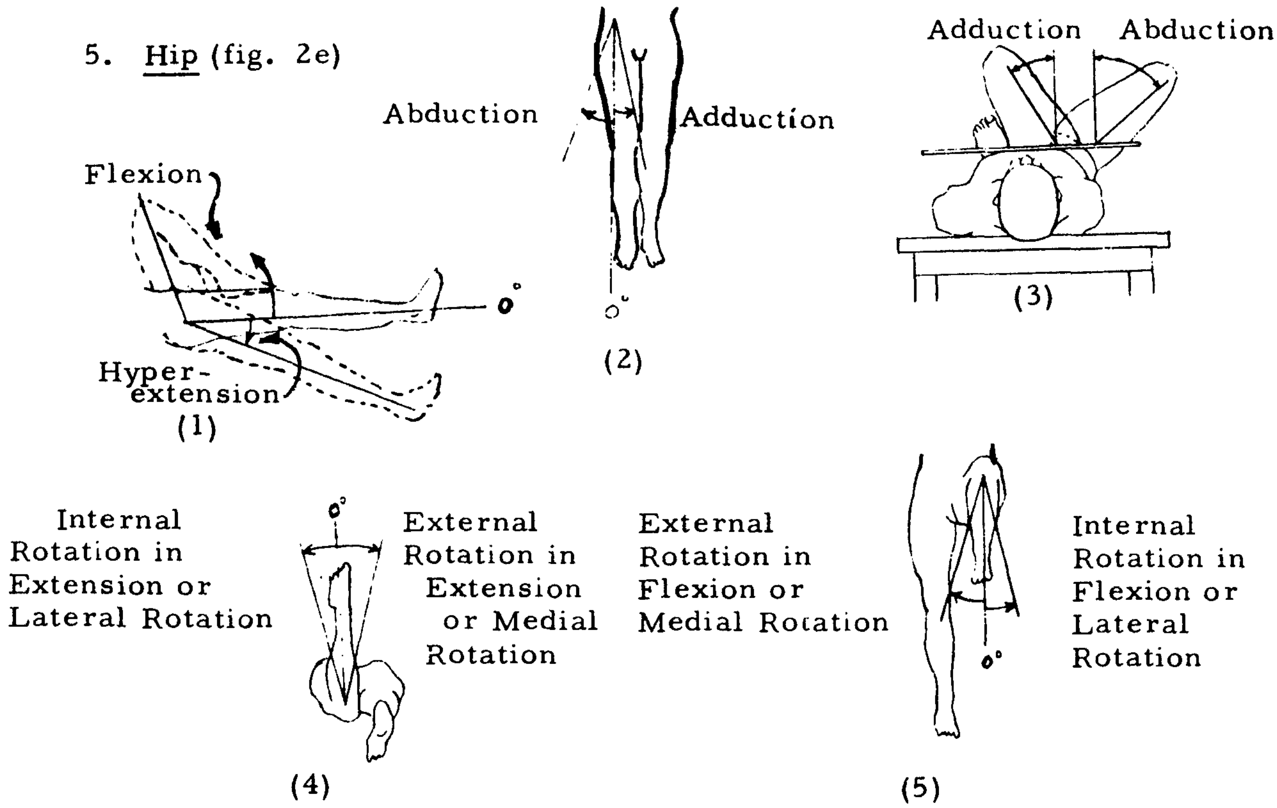


Figure 2e Motions at Hip

Flexion (1) (thigh moved upward in sagittal plane toward abdomen, allowing knee to flex)	Mean 120° (ref. 5) Mean 113°, S. D. 13° (ref. 6)
Hyperextension (1) (thigh moved downward and backward in sagittal plane)	Mean 45° (ref. 5)
Abduction (2) (in supine position, extended leg moved laterally)	Mean 45° (ref. 5)
Adduction (2) (same as above except thigh moved across the midline)	Mean 40° (ref. 5)
Abduction (3) (in supine position with knee bent 90° and heel in contact with table, knee and thigh turned outward)	Mean 53°, S. D. 12° (ref. 6)
Adduction (3) (same as above except knee and thigh turned toward midline of body)	Mean 31°, S. D. 12° (ref. 6)
Internal rotation in extension or lateral rotation (4) (body prone, knee flexed 90° and leg vertical, leg and foot rotated outward)	Mean 20° (ref. 5) Mean 34°, S. D. 10° (ref. 6)

External rotation in extension or medial rotation (4) (same as above except leg and foot rotated inward)	Mean 35° (ref. 5) Mean 39°, S. D. 10° (ref. 6)
Internal rotation in flexion or lateral rotation (5) (hip and knee each flexed 90°, leg and foot rotated outward)	Mean 30° (ref. 5) Mean 30°, S. D. 9° (ref. 6)
External rotation in flexion or medial rotation (5) (same as above except leg and foot rotated inward)	Mean 60° (ref. 5) Mean 31°, S. D. 9° (ref. 6)

6. Knee (fig. 2f)

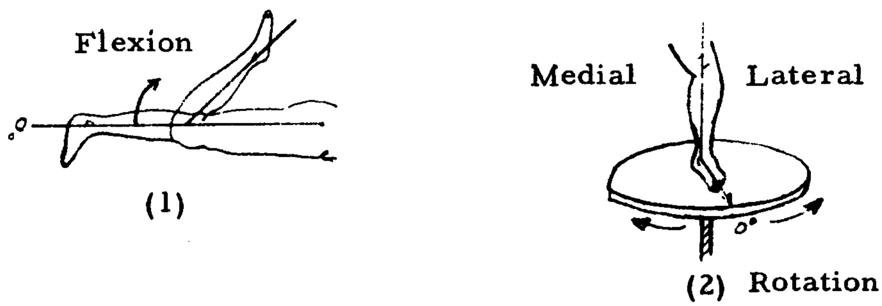
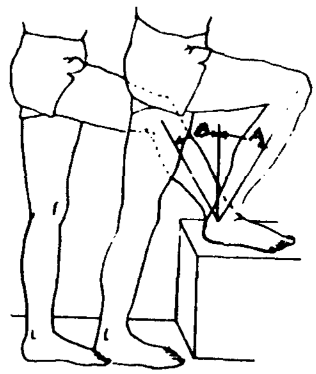


Figure 2f Motions at Knee

Flexion (1) (leg bent backward toward posteria surface of thigh)	
Prone position	Mean 135° (ref. 5) Mean 125°, S. D. 10° (ref. 6)
Standing position	Mean 113°, S. D. 13° (ref. 6)
Medial rotation (2) (subject has foot resting on turntable with ankle above the center, leg vertical, and thigh about 45° to vertical. Rotate leg and foot about vertical axis inwardly toward body center line)	Mean 35°, S. D. 12° (ref. 6)
Lateral rotation (2) (same as above except leg and foot rotated outward)	Mean 43°, S. D. 12° (ref. 6)

7. <u>Ankle</u> (fig. 2g)



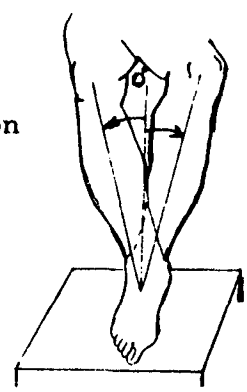
(A) Flexion or Dorsiflexion
(B) Extension or Plantar Flexion

Figure 2g Motions at Ankle

Flexion (A) (from thigh horizontal and foot flat on box with toes forward, leg moved forward and downward toward foot)	Mean 20° (ref. 5) Mean 35° S. D. 7° (ref. 6)
Extension (B) (same as above except leg moved backward and downward away from foot)	Mean 35° (ref. 5) Mean 38° S. D. 12° (ref. 6)

8. <u>Foot</u> (fig. 2h)

Eversion



Inversion

Figure 2h Motions at Foot

Inversion (knee flexed, foot on box with toes forward, knee and leg swung inward toward midline with foot sole kept horizontally on box)	Mean 24° S. D. 9° (ref. 6)
Eversion (same as above except knee and leg swung outward away from midline of body)	Mean 23° S. D. 7° (ref. 6)

The above data indicating the normal ranges of some of the human joints for certain particular motions of the limbs are useful in that they serve as a guide for estimating the joint range requirements of a Man Amplifier device. However, several factors revealed in this study of human joints are worth noting since they may present major problems affecting Man Amplifier component designs. One factor is that the instantaneous centers of rotation of human joints do not remain fixed at a point, i. e., the motion is not circular as would be obtained in a pin-centered joint. This means that, to avoid painful results, the man must have freedom to adjust his position in the exoskeletal structure to maintain his instantaneous centers in alignment with the exoskeletal joint axes. For most joints, the magnitude of the joint center shift is probably quite small; e. g., in reference 4, it is stated that, at the elbow joint, the cluster of instantaneous centers range over an area of about one-half-inch diameter. At the shoulder, however, the effective center of the composite system of joints between the trunk and humerus moves over a wide range of locations. The problem visualized here has to do with the sensors used to control the skeletal motions. If, because of the shifting about of his joint centers, the man must be able to move around inside the Man Amplifier structure, the sensors must be so designed as to permit this movement without producing signals which would result in unwanted forces or motions of the machine.

Another factor, similar in effect to that pointed out above, is that the motions permitted by single-degree-of-freedom-joints (e. g., elbow, ankle) are not truly planar; secondary, concomitant motions on axes perpendicular to the principal movement are induced. Thus, if a single pin joint is used at one of these places in the skeletal structure, these secondary or contingent movements of the subject will also result in relative motion between the man and machine and, hence, must be taken into consideration in the sensor design.

A third factor which should be noted is that motion ranges of some joints are dependent upon the positions of various body segments. For example, palmar flexion of the wrist is much less when the fingers are clenched in a fist than when the fingers are extended. Another example is the hip joint; permissible flexion of the hip is much less when the leg is in an extended position than when the knee is allowed to flex. It is possible, in cases such as these, that some sort of interlock system will be required to adjust the range limits of the joints according to the positions of other influencing members.

It is also important to note that there are appreciable differences in the motion ranges of various joints among individuals. If the Man Amplifier were designed with fixed joint range limits, then in the interests of safety, each limit should be no larger than that value which can be attained by a large percentage of personnel using the device. This would result in a man-machine system restricted to the motion capability of a (perhaps unrealistic) "least flexible" man and may seriously degrade the utility of the device. Alternatively, and at the expense of greater complexity, the motion limit of each joint could be made adjustable to match the capabilities of each individual.

It is clear that the human joint system is very complex and that it would be impractical, if not impossible, to construct a Man Amplifier skeletal structure which would duplicate all of the joints and motion ranges of the human body. On the other hand, it is quite unlikely that the entire human kinematic capability need be provided. Therefore, the problem is presented of determining those joint motions and their ranges which may be considered as minimal and essential to the satisfactory performance of the man-machine system without undue restriction. Some of these motions are obviously required (e.g. flexion of the shoulder, elbow, knee, hip), but the necessity of some of the others is more obscure and depends on how well the man can adapt himself, i.e., learn to do things, when denied the complete freedom of motion that he normally possesses.

As an illustration, consider the leg motions involved in the relatively simple (but certainly necessary) tasks of walking and turning a corner. In walking, the major motions of the lower extremities are flexion and hyperextension of the hip, flexion of the knee, plantar flexion and dorsiflexion of the ankle, and hyperextension of the toes. Freedom of motion at the hip, knee and ankle are probably necessary. Whether bending at the toes is essential is subject to question because a man can walk without this motion which allows him to rise up on the ball of the foot. How much influence this unnatural, restricted motion might have on the psychological well-being of the man and the degree to which it may impede the performance of his tasks is difficult to ascertain however. The same is true in turning a corner. Will it be necessary to incorporate a rotary motion capability at the hip joint to allow the foot and leg to twist with respect to the trunk in turning around, or will a simpler scheme such as pivots on the bottom of the feet suffice?

At the present time, decisions as to what joint-motion ranges should be provided in the Man Amplifier and whether some types of motions can be eliminated entirely cannot be made on purely rational grounds. Design decisions cannot be made without knowledge obtained from experiments in which movements of various body segments are inhibited by positive restraints.

Consideration must also be given to the dynamic performance characteristics of human beings before a suitable kinematic match between the man and the machine can be obtained. As shown later in the discussion of the servo system and power requirement studies, the angular-velocity requirements for the Man Amplifier joints are an essential factor in determining suitable types of servovalves and power supplies.

Little information on maximum attainable angular velocities for limb rotations about the various joint axes was found in a brief literature search. However, some of this needed information was determined from unpublished test data* shown plotted in figures 3 through 6. These curves show the measured time history of limb position, determined by means of high-speed photo-

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* Personal communication from Bennett, W. G., 2nd. Lt., USAF, Anthropology Section, Behavioral Sciences Laboratory, Aerospace Medical Division, Air Force Systems Command.

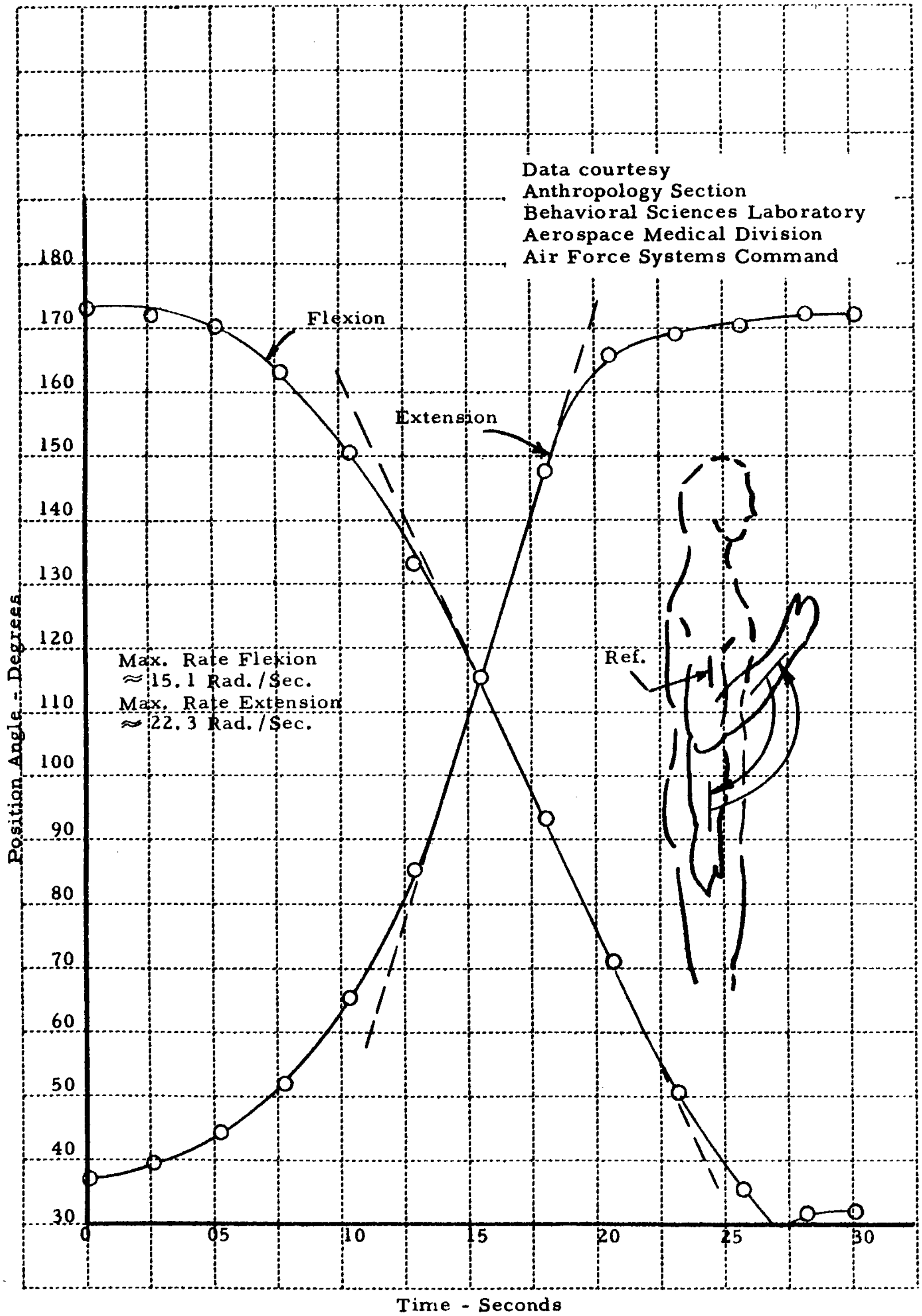

Figure 3 Maximum-Effort Flexion and Extension of Right Elbow

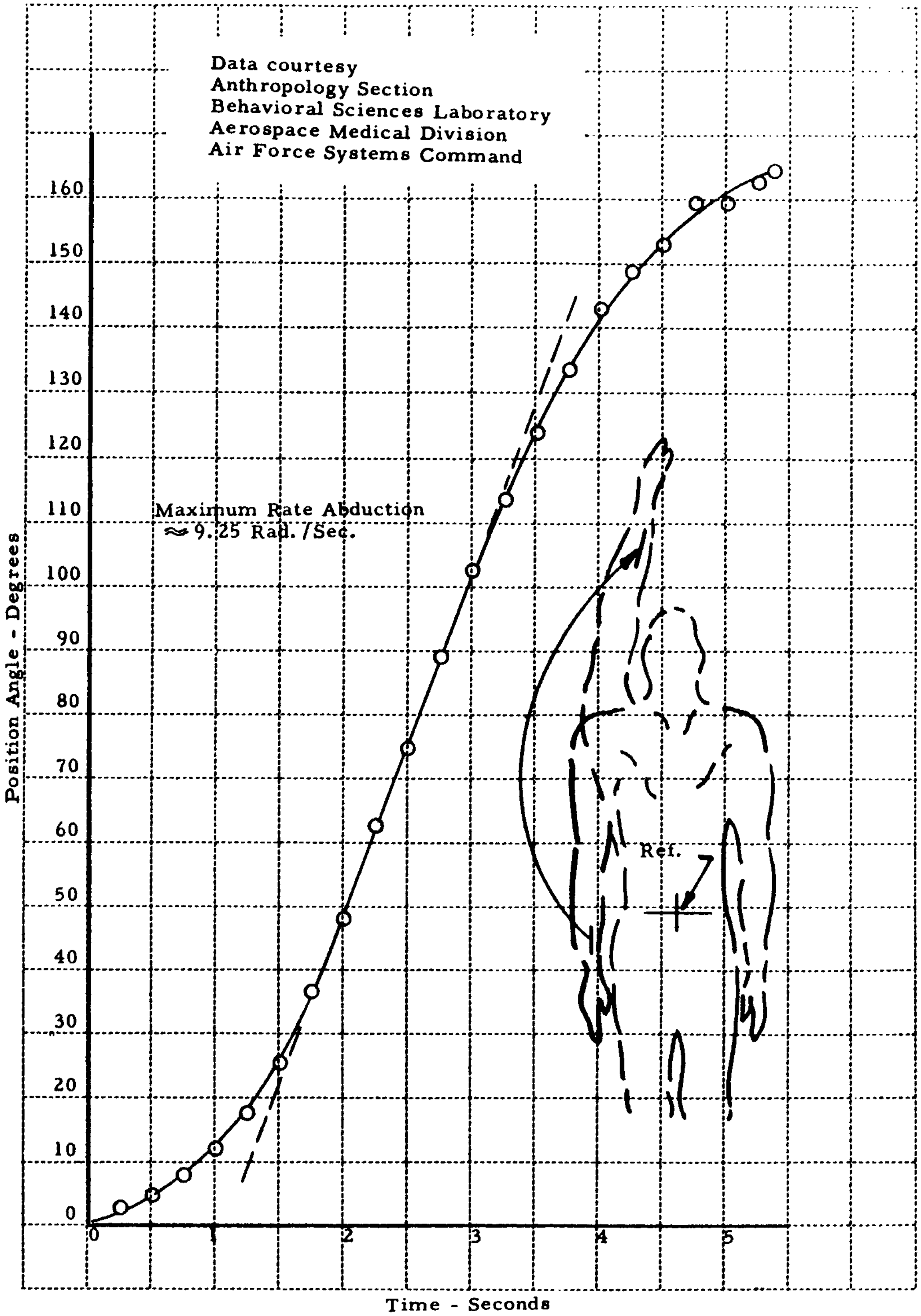

Figure 4 Maximum-Effort Abduction of Right Arm

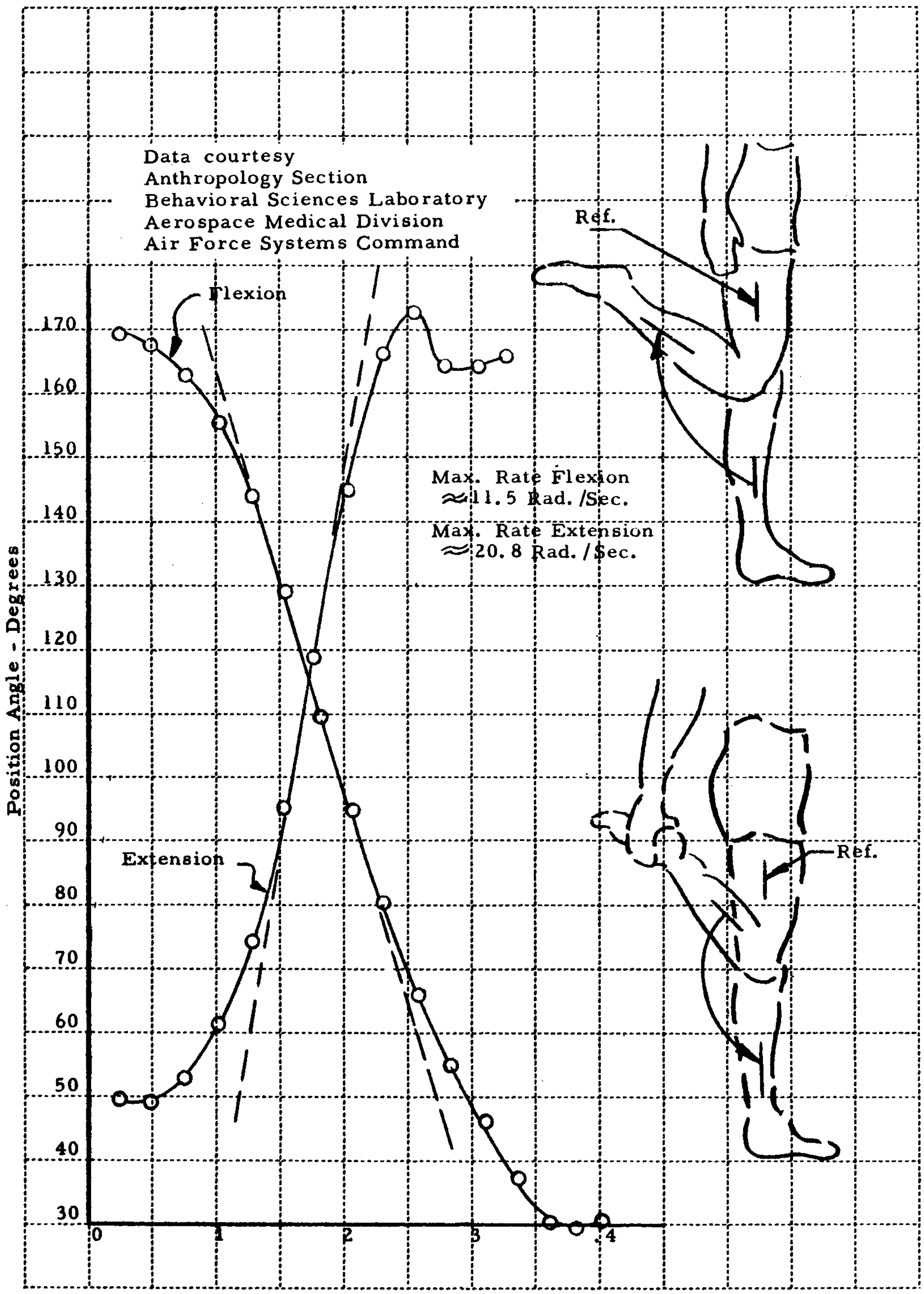

Figure 5 Maximum-Effort Flexion and Extension of Right Knee

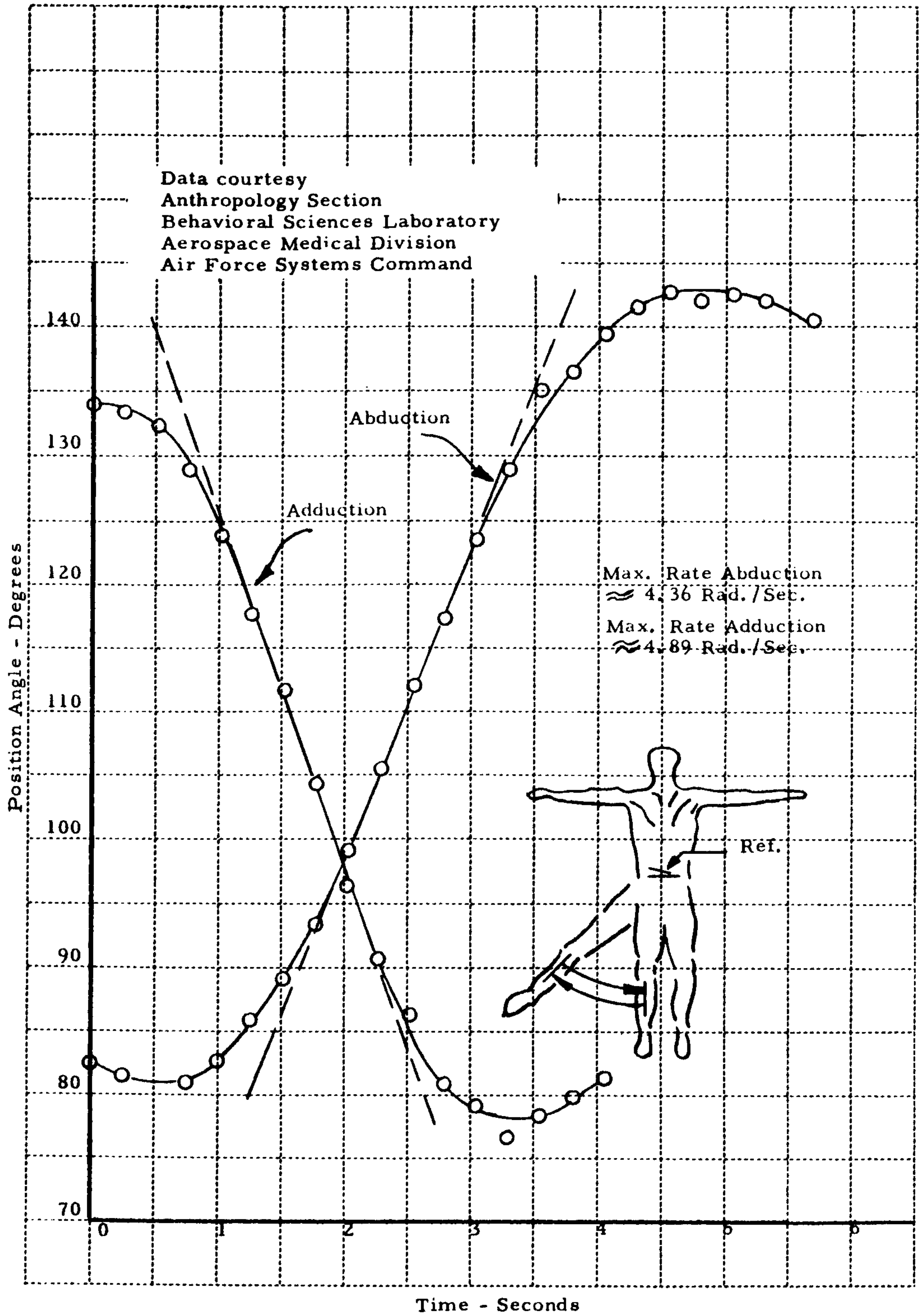

Figure 6 Maximum-Effort Abduction and Adduction of Right Leg

graphy, for one subject as he performs each motion at maximum effort. The maximum angular velocities achieved in these experiments, represented by the maximum slope of each curve, are indicated on the figures. These values (maximum rates for no load conditions) constitute upper limits for the angular motion rate requirements of joints to be provided in a Man Amplifier and serve to indicate the approximate joint angular velocities required if the device must necessarily follow the motions produced by reflex actions of the operator.

Preliminary Design-Structure and Weight

1. Design Criteria

One of the first problems encountered in a study of the structural and mechanical design aspects of the Man Amplifier is the definition of the critical design conditions for each member and component. Because a large number of possible design conditions could be postulated in terms of various combinations of configurations, loads, member orientations, tasks, etc., a determination of those conditions which form the most critical design requirements was clearly beyond the scope of a preliminary investigation. Consequently, it was necessary to limit the scope of the structural analyses to a consideration of only a small number of representative conditions, as set forth by the following assumed design criteria.

a. The tasks performed by the Man Amplifier consist of lifting, supporting (holding) and transporting a 1500 lb. payload.

b. The load factors for the several tasks are:

Task	Operating Load Factor	Limit Load Factor*
Lifting	1.1	2.2
Supporting	1.0	2.0
Transporting	1.1	2.2

c. The operating load factors tabulated above apply to the payload and the affected members of the Man Amplifier for a given design condition. In general, symmetry of load distribution is assumed except that,

1. The design load for each arm is 60% of the total arm load to account for possible unsymmetrical distribution of load while performing the various tasks.
2. The design load for each leg is 100% of the total leg load to account for probable load distribution during the transporting tasks.

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* Yield point of material not to be exceeded at limit load.

d. The total weight of the Man Amplifier system is assumed to be 510 lbs. consisting of

   1. 160 lb. operator

   2. 200 lb. structural "suit"

   3. 150 lb. power pack

   The assumed distribution of this weight and locations of joint centers and link lengths are shown in table III and figure 7 respectively. The data given in reference 7 were used as a guide in selecting these values.* For the structure, the ratio of the weight of each member to the total weight of the "suit" was assumed to be the same as that of each counterpart in the human body to the total body weight.

e. The investigation is limited to a two dimensional analysis, i. e., a consideration of motions, loads, moments, etc. produced with respect to joints having axes of rotation perpendicular to the sagittal (fore and aft) plane.

2. System Static Stability

A preliminary check of the Man Amplifier system performing the aforementioned tasks revealed that one of the limitations on its performance would stem from the inability to maintain static stability when handling heavy payloads. Thus, though the machine may be designed to provide greatly increased strength and power capability over the unaided human, it may not be possible to utilize this capability, in some instances, because the device does not provide the correspondingly increased stabilizing moment needed to counteract the overturning moment generated by heavy loads. This performance limitation is in contrast with that normally encountered by man, who is generally limited by a lack of muscular strength.

The static stability of the amplified man-payload combination is a function of the weight and distribution of the Man Amplifier system, the length and placement of the foot members, and the weight and center of gravity location of the payload being handled. Therefore, an investigation was made to determine the maximum load that the Man Amplifier could lift. Figure 8 illustrates the geometry that was assumed for this investigation. The orientation of the Man Amplifier as shown produces the maximum stabilizing moment in lifting the payload of arbitrarily selected shape and center of gravity location. Various lengths of the Man Amplifier foot member ranging between 10 and 18 inches were considered in the study.

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* Note that an extended forearm was assumed to account for mechanical hand extensions.

TABLE III

ASSUMED WEIGHT DISTRIBUTION OF MAN AMPLIFIER

Component	WEIGHT (pounds) Operator	Exoskeleton	Total
Head, Neck and Trunk	85	105	190
Upper Arms	11	14	25
Lower Arms and Hands	10	12	22
Upper Legs	35	43	78
Lower Legs and Feet	19	26	45
Subtotal	160	200	360
Power Supply		150	150
TOTAL		350	510

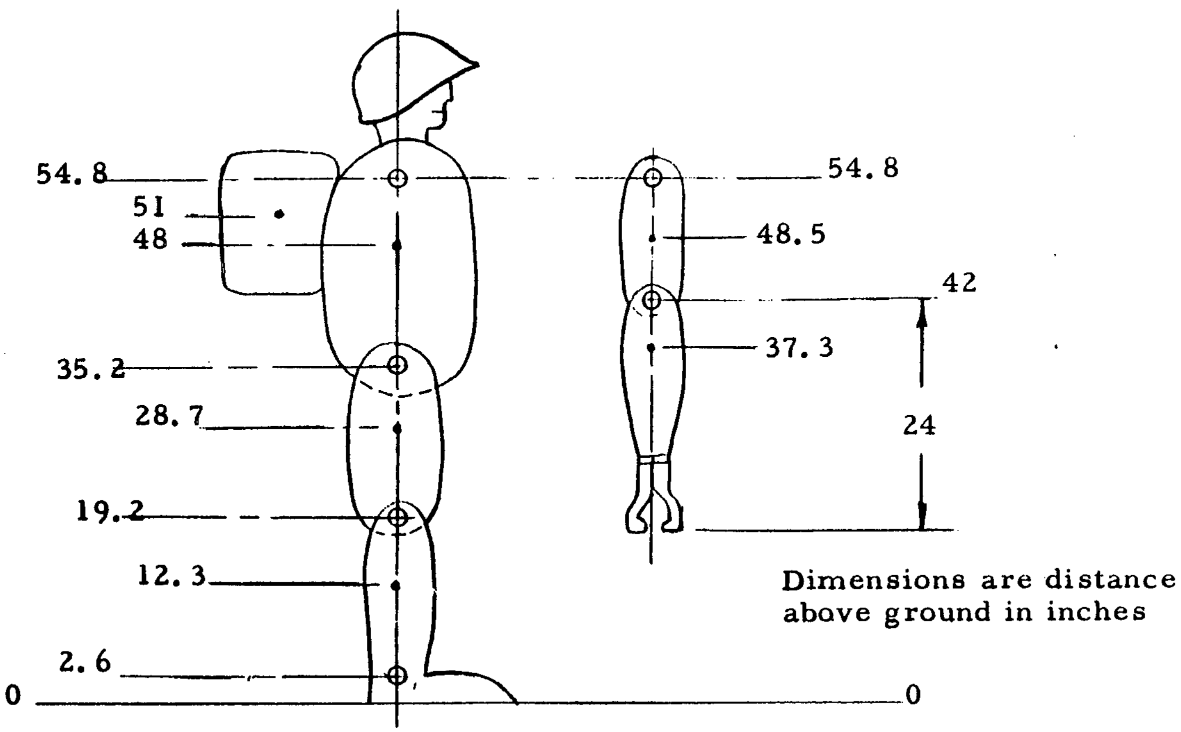

Figure 7 Assumed Locations of Man Amplifier Joints and Component Centers of Gravity

It was found that the maximum load that could be lifted by the Man Amplifier (for the assumed conditions) is approximately 1000 lbs. and is independent of the length of the foot member within the range of lengths considered. It should be noted, however, that the assumed posture of the Man Amplifier is unnatural and one which requires the operator to be supported by the Man Amplifier structure. Furthermore, other assumptions with regard to the weight of the operator and the machine, configuration, payload size and shape characteristics, etc. would result in different values for the maximum (stability limited) weight that could be lifted. For example, if the Man Amplifier could get its feet partially under the load, it would be capable of lifting the 1500 lb. maximum load previously assumed. (It should be emphasized also that the stability limitation discussed above applies to a lifting task and does not necessarily limit the ability of the Man Amplifier to exert forces of substantially greater magnitude in situations where static stability is not the controlling factor.)

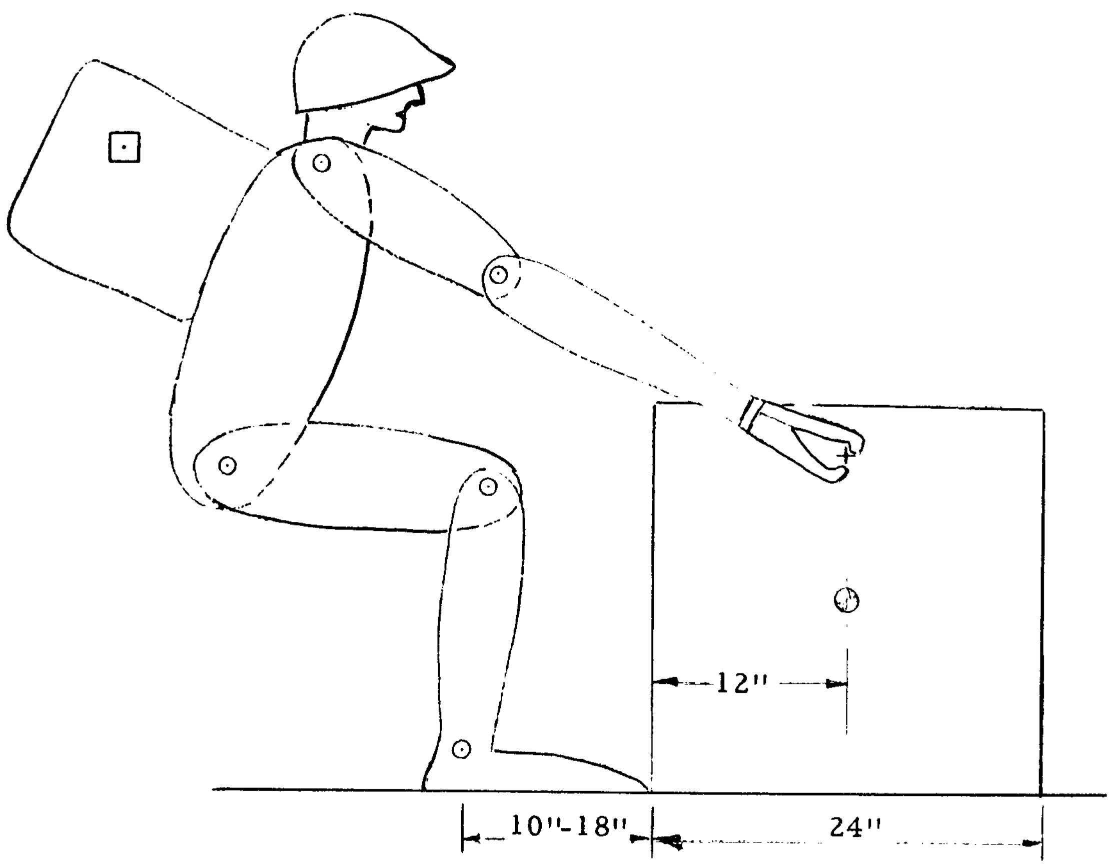

Figure 8 Assumed Geometry for Lifting Task

3. Structural Design Loads and Component Weights

A preliminary study was made to determine the maximum design loads and joint torque requirements for the exoskeletal structure. The Man Amplifier postures assumed for performing the lifting, supporting and transporting tasks are shown in figure 9. A summary of the limit design loads for each member are presented in table IV and are the maximum values occurring in each member for the three positions.

TABLE IV

SUMMARY OF LIMIT DESIGN LOADS FOR MAN AMPLIFIER MEMBERS

Member	Moment inch-lbs.	Shear lbs.
Forearm	47, 600	2020
Upper Arm	47, 600	2040
Body	82, 700	4080
Upper Leg	62, 000	4240
Lower Leg	31, 700	4290
Foot	31, 700	4290

No detail design effort was initiated during the course of this study in view of the lack of firm requirements. Accordingly, approximate structural characteristics were determined for the various Man Amplifier members by assuming simple cross-sectional shapes, realistic overall section dimensions, high strength aluminum alloy materials, and a maximum operating design stress equal to 50% of the material yield stress. On the basis of the above assumptions a simplified Man Amplifier configuration was generated which, space limitations notwithstanding, indicated that the strength of the exoskeleton members would not be a critical design consideration. The development of a design which will incorporate all of the essential degrees of limb freedoms in a workable configuration is viewed as the predominant problem and one which will require considerable ingenuity in the structural and mechanical design of the Man Amplifier.

The specification of the structural characteristics of the members treated in the above analysis permitted a more accurate weight estimate to be made than that assumed earlier in the arbitrary establishment of design criteria. This was done by alternatively assuming 7075-T6* and 6061-T6*

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* Alloy designation system for wrought aluminum by The Aluminum Association, 420 Lexington Avenue, New York 17, New York.

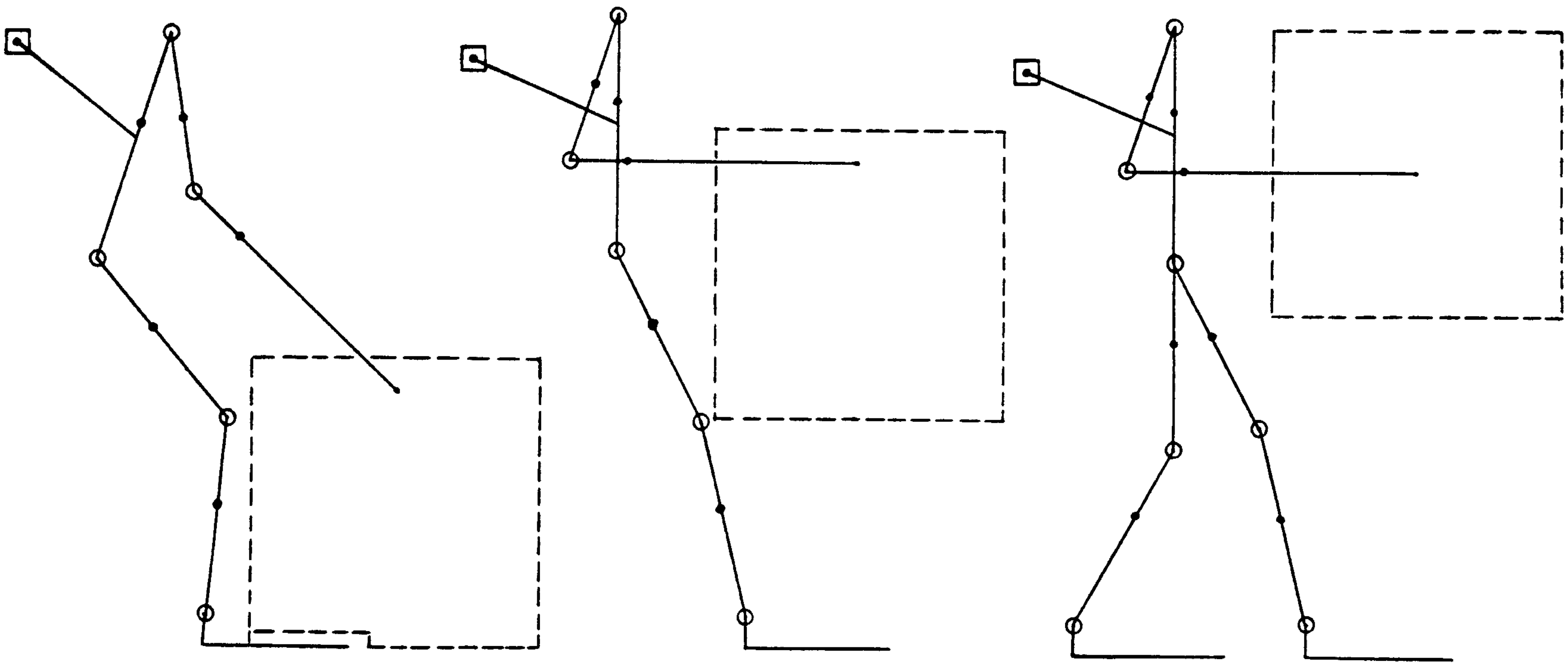

Figure 9 Assumed Man Amplifier Postures

aluminum alloy to be the basic structural material from which the Man Amplifier exoskeleton would be fabricated. The two-dimensional task analysis that resulted in the structural loads listed in table IV was expanded to include a three-dimensional, multi-axis configuration from which the structural weights of the primary exoskeleton members were determined. The resulting weights are presented in table V for the two structural materials considered. An estimate was then made of the weight of the structural fittings required to assemble the various skeleton members together with their respective joint servomotors to form the complete Man Amplifier structural assembly. This estimate was based primarily upon experience and intuition, with due consideration given to space limitations. Materials assumed for the fittings were identical to those of the corresponding basic structure.

Table V also includes an estimate for the total weight of the individual servomechanisms, plus the required sensors. The unit servo weight is based upon current state of the art. The total number of servos chosen is probably small for a completely developed system but is considered representative of an initial design.

Finally, a revised weight estimate is presented for the power supply required to operate the various servos and sensors of the system. This item is visualized as a portable or self-contained unit, with limited endurance, which forms an integral part of the Man Amplifier system. In terms of existing hardware, the estimated weight of the power supply is on the low side; however, it is believed to represent a figure that can be achieved by a rigorous developmental program.

TABLE V

ESTIMATED WEIGHT DIVISION OF MAN AMPLIFIER

Item	Weight (lbs.) for Specified Basic Structural Material	
	7075-T6	6061-T6
Structural Skeleton	62	117
Structural Fittings	73	137
16 Servos and Sensors	112	112
Integral Power Supply	100	100
Miscellaneous	20	20
TOTAL	367	486

A comparison of the total weights presented in the table indicates a definite weight advantage would be gained by employing the 7075-T6 material in the basic structural design of the Man Amplifier. There are, however, several additional considerations of perhaps equal or greater importance which bear on any final material selection. Although low weight is desirable to facilitate handling of the system, donning and removal of the exoskeleton by the operator, etc., a small weight tends to aggravate the static-stability problem discussed previously.

A significant advantage of the 6061-T6 alloy is the excellent weldability of this material. This capability will add substantial flexibility to the designer of the Man Amplifier who is faced with the task of incorporating elements embodying numerous degrees of rotational freedom in relatively confined areas. Additional considerations include the need for large torsional and bending stiffness in the basic structure in order to minimize inadvertent sensor actuation and similar related phenomena.

From the weight summations presented, it is estimated that a Man Amplifier system, as currently envisioned, would weigh somewhere between 525 and 650 lbs. for a 1500 lb. maximum payload capability. The estimates and calculations, based upon a necessarily limited scope of investigation, are believed to represent realistic values.

4. Summary of Structural-Design Problem Areas

Structural design considerations have resulted in the recognition of a number of specialized problems which require solution in the development of the Man Amplifier concept. Some of the major areas that will require comprehensive study are discussed briefly below.

a. System Static Stability - This aspect of the Man Amplifier system has been reviewed in some detail above. The problem will require additional study to permit the generation of an optimum configuration. Particular emphasis will have to be placed on a more comprehensive evaluation of tasks to be performed by the system.

b. Ease of System Activation - Briefly stated, "How does the operator don and doff a powered suit?" Does this operation require assistance from other personnel, and is this an assembly operation? Considerable design ingenuity will be required to minimize these and related problems.

c. Variation in Limb Size of Operator - The problem was discussed earlier and it was concluded that a certain amount of adjustment will be necessary in order to make the system practical. A possible solution would consist of individual structural limb members fabricated in several interchangeable lengths, each incorporating a small amount of adjustment, to accommodate in continuous fashion perhaps 80% of the male population. This procedure would require various subassembly operations but presumably does represent a feasible method for obtaining a large variation in exoskeleton size. Alternatively a single nominal length of limb with limited adjustment for a given pre-selected "sized" group, might prove to be an adequate solution of the sizing problem.

d. Required Flexibility of Motion - The degrees of freedom and range of motion with which the Man Amplifier system should be provided cannot be determined in a positive fashion without resorting to an experimental investigation. The investigation envisioned at this time would employ a nonamplifying exoskeleton equipped with numerous joints, each with "lock out" capability, such that the minimum amount and degree of flexibility needed to perform individual tasks could be assessed.

e. Material Selection - The general requirements for the structural exoskeleton of the Man Amplifier system are substantial strength, light weight, large bending and torsional stiffness to minimize deflections, limited bulk, and ease of fabrication. The possibility of employing materials other than 6061-T6, for example, should not be overlooked however, until the system requirements are more firmly established.

f. Safety - The problem of insuring safety of the operator under a myriad of operating conditions is a many faceted and complex one. Since much depends on the final design concept and method of operation, the problem of safety can be no more than broached at the present time. Suffice it to say that safety of the operator from all possible sources of danger, both internal and external to the Man Amplifier system, should remain paramount throughout the design task. Consideration must be given to limiting motions of powered members so as not to exceed the operator's motion range, isolation from power supply operation, protection from electrical shock, emergency egress, etc.

Servo System and Power Requirements

Among the servomechanism performance requirements for each powered joint of the Man Amplifier are: (1) the maximum torque, (2) the maximum angular velocity, and (3) the maximum power. Preliminary estimates of the joint torque and power requirements were obtained by considering the same tasks assumed for the Man Amplifier in the preceding section. The maximum torque occurring at each joint was determined for the positions of the Man Amplifier shown in figure 9. However, the power requirements are a function of both the torque and angular rate at any instant; hence, a time measurement must be introduced.

The time interval for the lifting task, in which the Man Amplifier posture changes from the initial position shown in figure 9(a) to the final position shown in figure 9(b), was obtained by assuming the load is lifted at a constant vertical velocity of one foot per second. Time was introduced in the transporting task by assuming a walking rate of 60 steps per minute. The average angular rate for each joint was computed by measuring the approximate angular range through which each joint moved during each task. Using these values of average angular rate, the maximum products of joint angular rate and simultaneous required torque were determined for the tasks indicated. A summary of the individual joint servo design requirements as determined above, along with estimated maximum no-load angular velocity requirements, is shown in table VI.

In this table it should be recognized that the horsepower values shown are the highest of the average horsepower requirement for each joint in performing the two tasks considered and hence do not all occur simultaneously. Therefore, the total power requirement is not a summation of the individual joint requirements listed.

An indication of the total power required may be obtained by a consideration of the lifting task, for example. The pertinent factors for each joint are listed in table VII.

For the shoulder joint, the load produces a torque tending to rotate the joint in the direction required by the task and therefore may not represent a true servo requirement, depending upon the design of the system. Because the sum of the above joint powers represents only one side of the Man Amplifier, the total servo output power required is about 9 horsepower. It should be noted that this power was derived on the basis of an average torque, a constant average angular rate for each joint, and also under the assumption that all joints moved simultaneously at their respective average rates during the entire time interval in lifting the load. However, the numerous joints permit the same task to be performed in many ways and other assumptions for the manner in which the various joints moved would yield different results for both the individual joint and total power requirements. Although the assumed Man Amplifier limb motions are not truly representative of the actual conditions, they nevertheless provide a basis for obtaining an order-of-magnitude result for the required joint servo power output.

In the design of the servomechanisms required to power the amplified man, considerable attention must be given to the problem of maximizing efficiency. Because of the demands placed upon the prime mover and fuel supply, the problem is particularly critical when the power supply is an integral part of the Man Amplifier package and long-term operation is desired.

TABLE VI

SUMMARY OF JOINT SERVO DESIGN REQUIREMENTS

Joint	Torque (lb-ft)	Angular Rate at Indicated Torque (Rad/sec)	Horsepower	Estimated Max. Angular Rate at no load (Rad/sec)
Elbow	2000	0	----	22.3*
	1400	.75	1.91	
Shoulder	1650	0	----	10.5***
	1600	.47	1.37	
Hip	2600	.78	3.69	6.1***
Knee	1300	2.62	6.20	11.5**
Ankle	950	.20	.35	2.6***

* Figure 3
** Figure 5
*** Estimated for running subject

TABLE VII

SUMMARY OF JOINT SERVO REQUIREMENTS FOR LIFTING TASK

Joint	Average Torque (ft-lbs)	Average Angular Rate (Rad/sec)	Horsepower (one side only)
Elbow	1340	.75	1.83
Shoulder	1280	.47	1.09
Hip	1450	.32	.84
Knee	680	.35	.43
Ankle	600	.20	.22
			$\sum$ = 4.41

Although other types of control systems may be practical, or more advantageous, hydraulic servos were considered in this investigation because they provide excellent torque-speed characteristics with maximum torque available at stall. Further, stall operation can be maintained indefinitely without deleterious effects.

In general, hydraulic servomechanisms utilize constant pressure power supplies and flow metering servovalves which control the fluid flow to an actuator (or motor) as some function of the servo error (i. e. the difference between the desired and actual responses). Typical flow-pressure characteristics of a constant-pressure hydraulic power supply are shown in figure 10. In power supplies of this type, a variable displacement hydraulic pump is generally used to provide constant pressure at any flow within the regulation range. The pump displacement is controlled by a pressure sensing device such that supply flow is provided according to the requirements of the load. If the required load flow were zero, for example, the pressure sensor would modify the pump displacement so no flow would result (i. e., the pump stroked volume would become zero). These devices are designed to use constant-speed prime-mover drives.

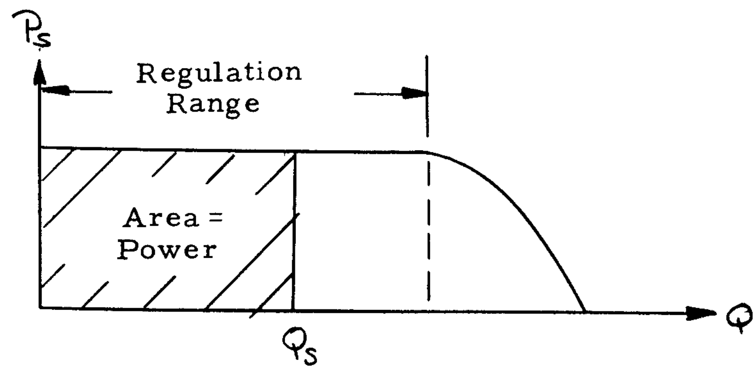

Figure 10 Characteristics of a Typical Hydraulic Power Supply

Note that the power output of the pump is the area beneath the curve from the origin to the flow supplied at any time. That is:

$$HP_s = \frac{1}{6600}\int_0^{Q_s} P_s \, dQ = \frac{P_s Q_s}{6600} \text{; if } P_s \text{ is constant.} \qquad (1)$$

It is desired to maximize the percentage of this power that reaches the servomechanism load.

In a typical hydraulic servovalve, the output flow is proportional to the valve opening and the square-root of the valve pressure drop. The valve pressure drop, in turn, is the difference between load pressure and supply pressure. Hence,

$$Q_V = k\chi_E \sqrt{P_S - P_L} \tag{2}$$

Equation (2) can be plotted to indicate the form of the valve flow-pressure response to valve spool displacement, as shown in figure 11.

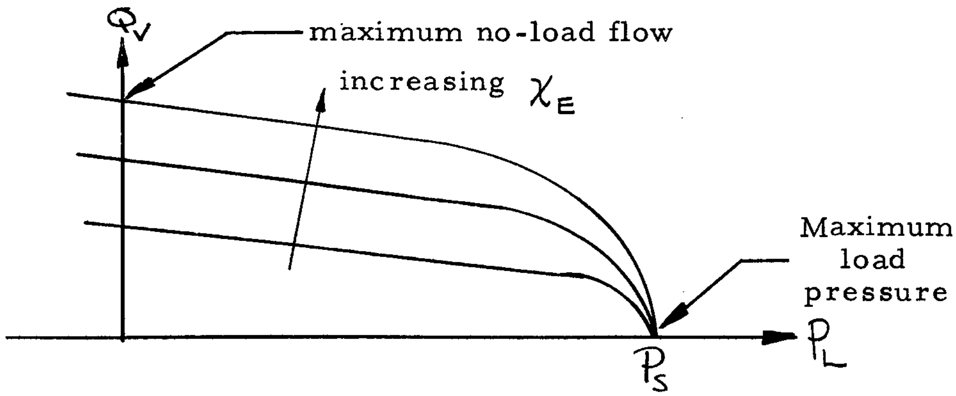

Figure 11 Servovalve Flow-Pressure Response

It is apparent from an examination of this figure that the servovalve flow is a function of pressure for a constant spool displacement. When the servovalve is "wide open" and the actuator is stationary, no flow will result but the load pressure will be maximum. Conversely, if all actuator loads are removed, the actuator velocity will be limited by the flow that results when full system pressure is dropped across the servovalve.

For a single servo connected to the power supply in a closed system, the servovalve flow must equal the power supply flow. That is,

$$Q_V = Q_S$$

The power delivered to the load by the power supply is therefore given by Equation (3).

$$HP_L = \frac{P_L Q_V}{6600} = \frac{P_L Q_S}{6600} \tag{3}$$

The efficiency of power transfer from the supply to the load is the ratio of load power to supply power, as indicated by Equation (4). This efficiency

$$\eta = \frac{HP_L}{HP_S} = \frac{P_L}{P_S} \qquad (4)$$

is 100% when the load and supply pressures are equal, but the load velocity (oil flow) is zero for this condition as shown in figure 11. Consequently, no power is delivered to the load. In other words, maximum efficiency occurs when zero power is delivered to the load and zero power is taken from the power supply.

When the load force is zero, no pressure is applied to the actuator and, if the servovalve is then wide open, the maximum load velocity is achieved. Again no power is delivered to the load, but in this case the efficiency is zero. All of the input power is dissipated in the servovalve by heating of the hydraulic fluid. In this case the maximum possible power is drawn from the power supply and none of it is delivered to the load. This is clearly undesirable.

Because the load power is zero when either the load velocity or the load force is zero, maximum power transfer must occur at some intermediate point where neither force nor velocity are zero. This point occurs when the product of P_L and Q_V is a maximum. It can be shown that maximum power transfer occurs when:

$$P_L = \frac{2}{3} P_S$$

and

$$Q = \frac{1}{\sqrt{3}} Q_{V\,MAX}. \qquad (5)$$

It follows that the efficiency at maximum power transfer is 66.7%. Figure 12, below, illustrates the major points brought out in the above discussion.

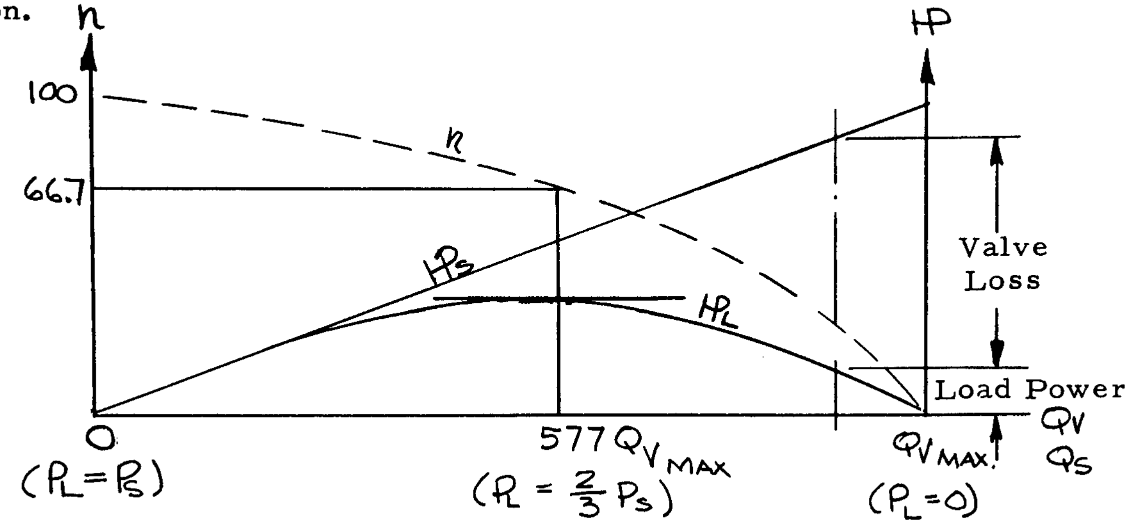

Figure 12 Power and Efficiency Versus Flow

As noted above, power lost in the transmittal of power from the supply to the load is almost entirely dissipated in the servovalve. If the power supply could be coupled directly to the load actuator, without an intermediate servo-valve, and if the power supply fluid flow could be modulated as a function of the servoloop error, the efficiency could be made to approach 100% at all load-force, load-velocity conditions. A direct-coupled power supply of this sort would necessarily have equal load and supply pressures. One method that could be used would employ a variable displacement pump to drive a fixed displacement motor or actuator and a means to modify the pump displacement with the feedback signal in order to reduce servoloop error to zero. Such a pump would have flow-pressure characteristics similar to the curves of figure 13.

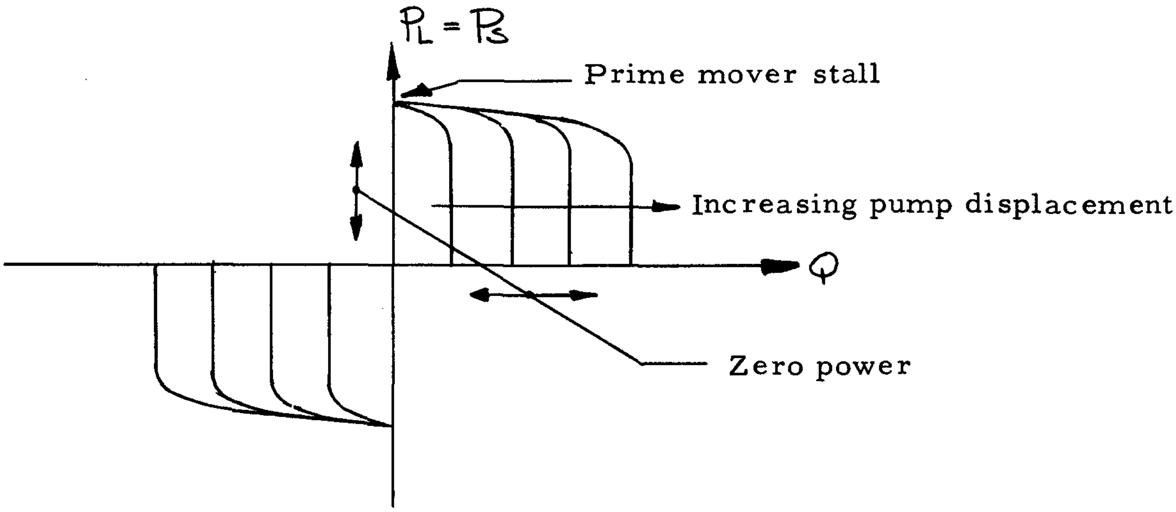

Figure 13 Characteristics of a Flow-Modulated Pump

As indicated in this figure, flow supplied at zero pressure, or pressure supplied at zero flow both result in zero power transfer and no power input to the pump (neglecting friction and windage losses). The pump, assumed to be driven at a constant speed, would probably have its stroked volume modulated by a small auxiliary servo requiring 5 to 10% of the total main servo power in order to achieve adequate response times (ref. 8).

The conditions shown in table VI were used as a basis for determining the practicality and availability of contemporary hydraulic equipment as applied to contemplated Man Amplifier control system designs.

Assuming the power values shown are matched at the conditions of maximum power transfer, it is concluded that conventional valve-controlled servos are not capable of providing the maximum no-load angular velocities specified. This conclusion derives from consideration of figure 14.

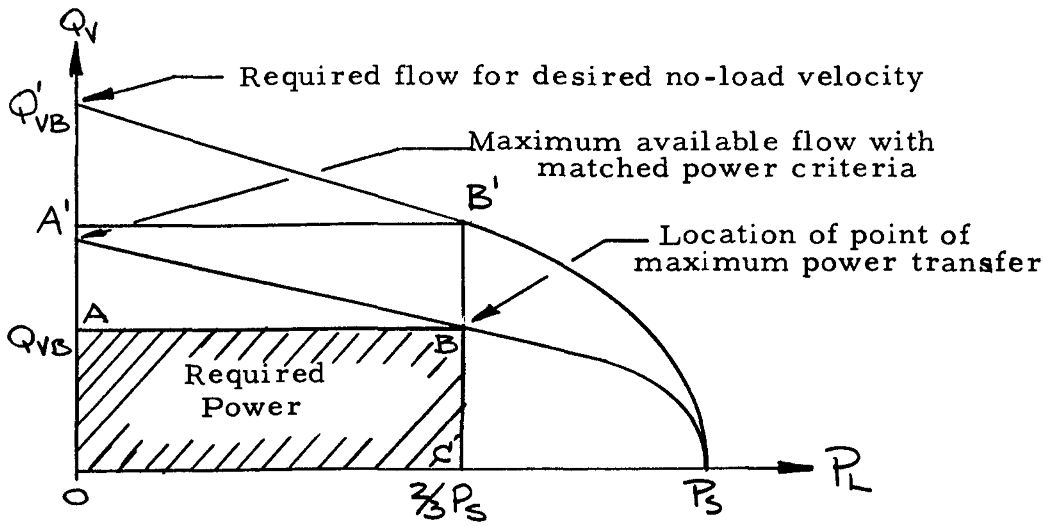

Figure 14 Illustration of Servovalve Power and Velocity Matching Problem

In this diagram the required power is represented by the area O A B C. Location of the maximum power point at B requires a limit on the no-load flow at Q_{VB} , which is less than the maximum no-load flow (Q'_{VB}) required to achieve the desired no-load velocity. If, on the other hand, a servovalve capable of supplying the flow required for the desired no-load velocity is assumed, the available load power will be O A' B' C, which exceeds the required power, O A B C. It is apparent that the two design criteria conflict, such that either one criteria or the other must be used. In the discussion that follows, the design solution is based on matching power requirements, and accepting the reduced no-load velocity that results.

Currently available rotary hydraulic servomotors, plus appropriate gearing, were investigated because of their applicability for the relatively large angular displacements that will be required for the Man Amplifier system. The hydraulic supply pressure and flow required to achieve the previously specified load torque and velocities (see table VI) are given by Equations (6).

$$\left.\begin{aligned} P_S &= \frac{2\pi T_L}{\sigma D_m N} \\ Q_S = Q_V &= \frac{D_m N \Theta_L}{2\pi} \end{aligned}\right\} \qquad (6)$$

On recalling that $Q_V)_{max.} = \sqrt{3} Q_V)_{P_L = 2/3 P_S}$, (Equation (5)) and assuming (1) a hydraulic supply pressure of 3000 psi and (2) a servomotor volumetric efficiency of 1.0, the factor $D_m N$ can be computed from the first of Equations (6). The no-load flow is computed from the second equation. A catalog of available servomotors (Vickers, Inc., Bulletin No. A-5205) was consulted to get representative values of servomotor displacements, taking into account maximum speed capability and physical size of the servomotor. The required gear ratios were computed after the servomotors were selected. Table VIII summarizes the findings and, on comparison with the conditions listed in table VI, shows that contemporary hardware can satisfy the required torque and power but not the no-load angular rates unless considerable additional power is made available.

If all joint servos simultaneously provided the load power indicated in table VIII, the total load power would be 26.5 horsepower. This, in turn, would require the output of the hydraulic power supply to be a minimum of 40 horsepower, if valve-controlled servos (operating at the point of maximum power transfer) were used. The hydraulic power required for all servos to move at maximum velocity and no-load would be 69 horsepower. Because the power necessary to achieve velocity-limited operation is greater than the power required to operate at the point of maximum power transfer, it is necessary to design the hydraulic power supply to provide the cumulative flow required to simultaneously operate several or all of the servos in the velocity-limited condition. This loading "duty cycle" must be carefully controlled to prevent a general loss in supply pressure due to an overloaded power supply; accordingly, the problems involved with the sudden loss of supply pressure must be minimized through careful design.

Supplementary Considerations

Certain problems pertaining to the task of assessing the feasibility of the Man Amplifier concept could be given only superficial consideration in this preliminary study. Some of these problems and uncertainties, believed to be in need of more detailed study, are briefly discussed in the following paragraphs.

One of the more difficult problems to be solved in the development and implementation of the Man Amplifier concept is that of specifying the sensors that form the communicative links between the operator and the mechanism he controls. One facet of the sensor problem has been discussed previously, i.e., the need to allow for some freedom of movement of the operator relative to the exoskeletal structure without producing signals that would result in undesired motions or forces by the machine. There are, however, many other aspects to the sensor problem. It is necessary to determine (1) what must be sensed (e.g. position, force, velocity, acceleration), and (2) the type of device (i.e., its physical form and principle of operation) that will provide the required command input and feedback signals in each servomechanism control loop. Sensor location and orientation, dimensional changes of the body due to muscle action, the possible hampering effects of clothing worn by the operator, and the signal processing needed in coupling the sensor outputs to the appropriate joint servos, are additional facets of the overall sensor problem.

TABLE VIII

SUMMARY OF SERVOMOTOR AND GEARING REQUIREMENTS TO SATISFY POWER REQUIREMENTS OF TABLE VI

Servo	No-Load Rate (rad/sec)	Full-Load Torque (ft-lb)	Maximum Power Output (horsepower)	Servovalve Size (in^3/sec)	Servomotor* Displacement (in^3/rev)	Servomotor Weight (lb)	Gear Ratio
Ankle	0.35	1420	0.35	2.0	0.128 (1)	4.5	264
Knee	4.54	1990	6.3	36.1	0.367 (2)	4.9	136
Hip	1.36	3900	3.7	21.2	0.367 (2)	4.9	264
Shoulder	0.82	2360	1.36	7.8	0.159 (3)	4.5	374
Elbow	0.75	2110	1.91	11.0	0.190 (4)	4.9	279

* From Vickers, Inc., Bulletin A-5205

(1) MF 36 3907-20Z
(2) MF 36 3909-30Z-12
(3) MF 36 3907-25Z
(4) MF 36 3909-15Z-12

Serious servoloop design problems can occur as a result of compliance or compressibility of the structure and servomotor working fluid coupled with lightly damped high-inertia loads. These physical properties of the servomechanism elements almost always limit the performance (transient and frequency response) of the servo because they contribute to unstable servo operation. Because of this instability, the closed-loop gains required for desired servo performance cannot usually be realized without stability augmentation. Augmentation techniques such as dynamic pressure feedback or transient-flow stabilization are usually effective means for achieving stable servo operation.* The possibilities of instability when servos are cascaded (as in the shoulder-elbow-wrist servos, for example) are greatly increased because each servo provides an "active" load on the others. This load consists of (1) the inertia of the servo mass and (2) the reaction forces and moments that result from the motions of the servo. The combined system could be unstable even if each individual servo was stable when considered alone.

During the process of performing the analyses described herein, many points and questions were raised that have been helpful in visualizing and planning the additional feasibility and research studies that need to be made. To a large extent, many of these questions have been in the area of man-machine relationships. For example, the following questions have been raised:

How well can the Man Amplifier, which responds to signals derived from sensors located at discrete points on the operator, be controlled and made to feel natural to the operator who receives kinesthetic feedback information from numerous sources distributed throughout his entire body? What should the force amplification (machine force/operator force) at each joint be and should it be adjustable to correspond with human strength variations as a function of the limb position? Will "feel" be natural if the amplification is not the same for each joint? To obtain complete and satisfactory answers to questions such as these, a substantial amount of research, both analytical and experimental, will be required.

* See Appendix A for an analysis of compliance-inertia instability in the CAL Elbow-Joint servo.

EXPERIMENTAL INVESTIGATION USING THE CAL ELBOW-JOINT AMPLIFIER

Summary

Man-machine compatibility in the tracking of a random position variable was studied experimentally. A hydromechanical position servo, originally constructed at CAL as a demonstration elbow-joint power amplifier, was used to provide power boost to the efforts of human subjects. As a portion of the program, various stabilization techniques were applied to the servo, both theoretically and in practice, to achieve desired transient and frequency response characteristics at all values of load inertia. With the servo stabilized, the frequency response characteristics were measured and compared with theoretical results, thus providing quantitative information on the servo behavior for correlation with the results of the tracking test. The servo analysis and frequency response data are presented in the appendices.

A tracking-test error analysis indicated that, without power boost, tracking error increases slightly, but not significantly, with increased load mass, until the subject fatigues. As fatigue mounts, the tracking error increases rapidly. With power boost, the tracking error was independent of load mass within the capacity range of the servo. Further, the tracking error with power boost for all loads employed was equal to the tracking error without power boost at no load; hence, the errors were due to the performance limitations of the subjects and not to the characteristics of the servo.

Description of the CAL Elbow-Joint Amplifier

The CAL elbow-joint amplifier is a hydromechanical follow-up servo designed to maintain a one-to-one position correspondence between a sensing arm (input) and a power arm (output) fastened to a load. Figure 15 is a schematic representation of the unit that was constructed. Referring to the diagram, application of a torque to the sensing arm, pivoted at point "0", changes the sensing-arm position. This movement causes the adjustable cam, attached to the sensing arm, to displace the servovalve spool. It is apparent that by varying the shape of the cam, the magnitude of valve-spool displacement per unit angle change between the sensing and power arms can be adjusted.

Displacement of the valve spool results in a flow of hydraulic fluid proportional to the valve spool displacement and the square-root of the valve pressure drop. This flow is directed into the actuating strut, on the appropriate end of the double-acting piston, forcing the power arm to move in the direction of the input arm motion until their relative displacement is zero. This action returns the valve spool to the null position and the system to static equilibrium. Note that the spool can be in the null position only if the power-arm angle is the same as the sensing-arm angle, i. e., the error is zero. Consequently, this servo is commonly called a zero position-error servo, a single-integration position servo, or a type 1 position servo. The servo converts position inputs to position outputs regardless of the load force, within the force-generating capabilities of the servo. No force-feel is provided to the operator in the CAL servo.

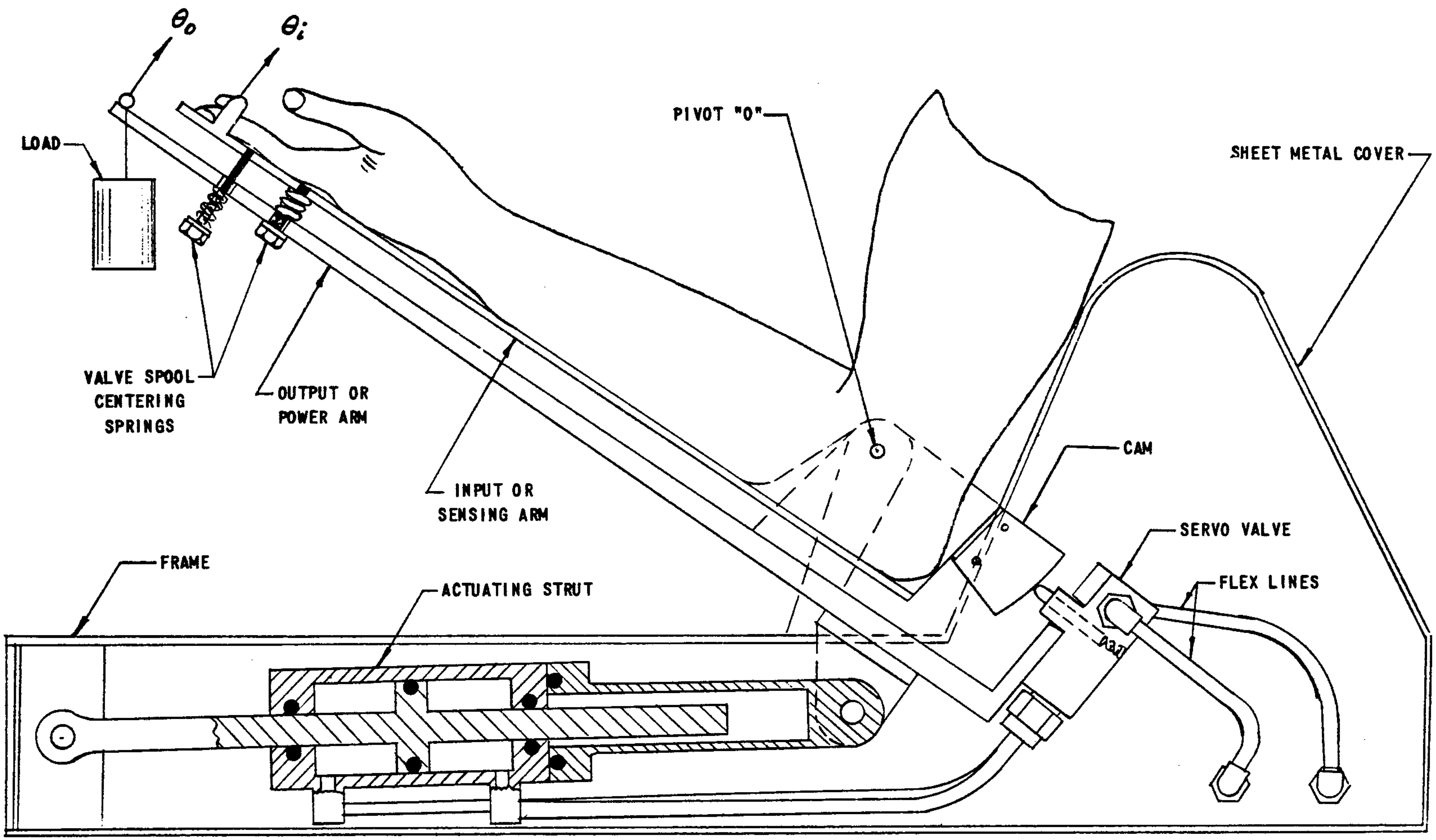

Figure 15 CAL Elbow-Joint Position Servo

An analysis of servo performance, as measured by steady-state sinusoidal frequency response tests, and a listing of the servoloop parameter numerical values can be found in the appendices.

Experimental Tracking Tests

A limited preliminary indication of the validity of the Man Amplifier concept was obtained from results of position-tracking tests in which the CAL elbow-joint amplifier was employed. In these tests, the input to the human operator was the viewed position of a pointer moving in a circular arc just beyond the end of the servo power arm. The subjects attempted to make a second pointer, attached to the end of the power arm, follow the input-pointer motions ("pursuit" tracking). To do this, the operator visually detected any error (viz. misalignment of the two pointers) and tried to minimize it by making the appropriate corrective motions of the servo input arm which, in turn, would cause the power-arm pointer to be properly positioned.

The input-pointer motions were produced by an electro-mechanical instrument servo coupled to the pointer by a cord and pulley arrangement as shown in figure 16. Amplified signals from an electronic noise generator, which produced constant spectral density or "white" noise in the 0-30 cps band of frequencies, drove the instrument servo. Frequencies above 1 or 2 cps would cause large tracking errors because of the relatively slow operator response. On the other hand, frequencies below .01 cps would contaminate a relatively short duration test by appearing as a very low-frequency drift, or bias, on the pointer random motions. Because of these factors, the noise generator output was filtered to reduce both the high- and low-frequency motions of the input pointer.

The amplitude ratio of the noise filter versus frequency was established by a first-order lead network having a corner frequency at .0133 cps followed by a second-order lag network having a .10 cps natural frequency and a damping ratio of 0.70. Both the asymptotic and the actual response characteristics of the filter are presented in figure 17.

A potentiometer fastened to the power arm of the CAL servo measured the response of the servo to the control motions of the operator. The random motion of the input pointer (restricted to excursions ± 30° from the center of the sector) was measured by means of a potentiometer mounted on the instrument-servo shaft. Because its pass band was much greater than that of the noise filter, the dynamic-response limitations of the instrument servo had little effect on the input-pointer response to the noise generator signals.

The CAL servo frequency response depends on the load inertia as shown in Appendix B, figures 23 and 26. With no load, the servo response was uniform to beyond 30 cps, but a 50 lb load reduced the bandwidth to about 7 cps with poor damping around the resonant frequency. However, this large change in the dynamic response of the elbow amplifier was not expected to influence the tracking accuracy because the input frequency content was limited to the unity amplitude-ratio range of the servo response.

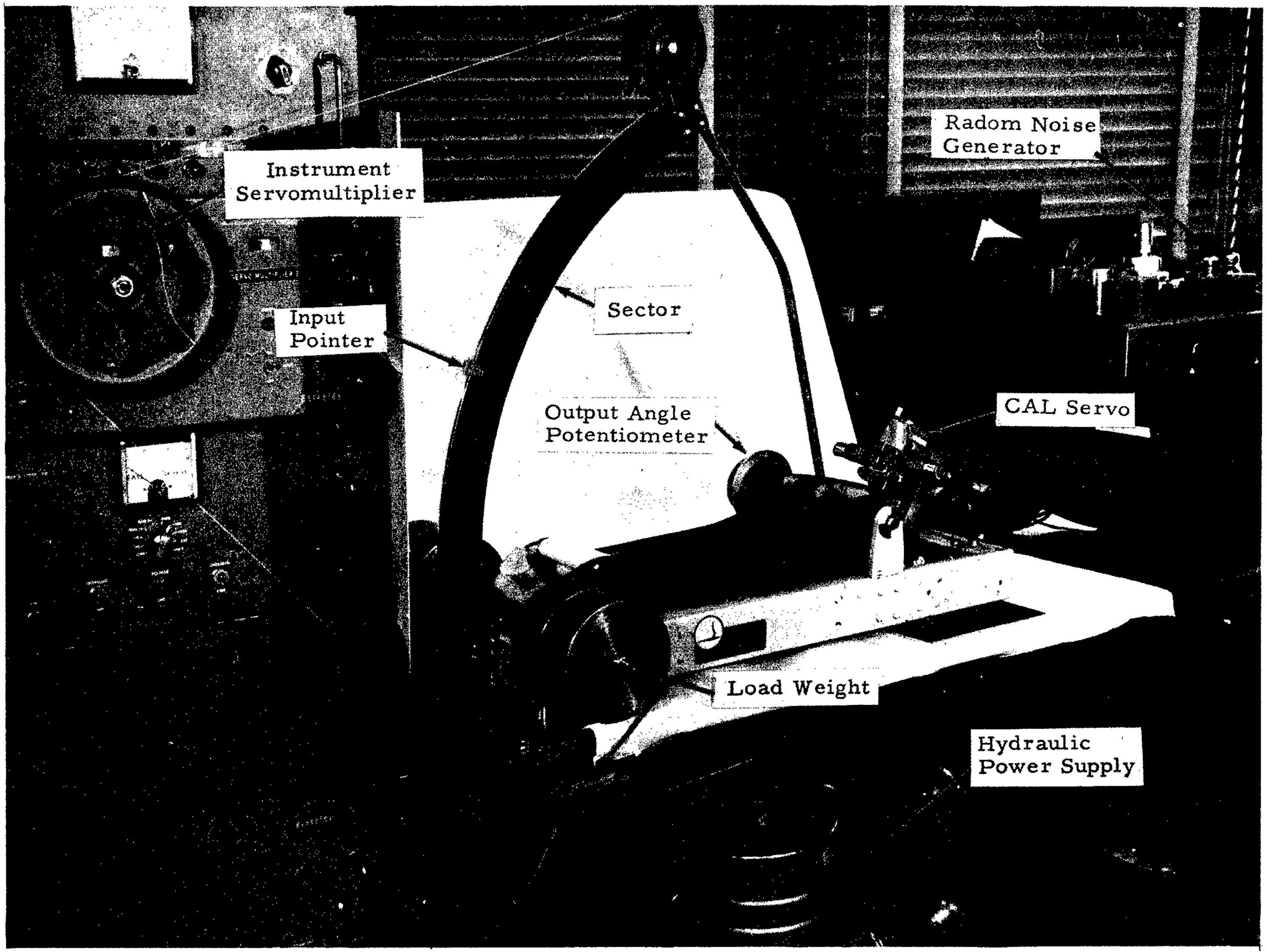

Figure 16 TEST SETUP FOR POSITION TRACKING EXPERIMENTS

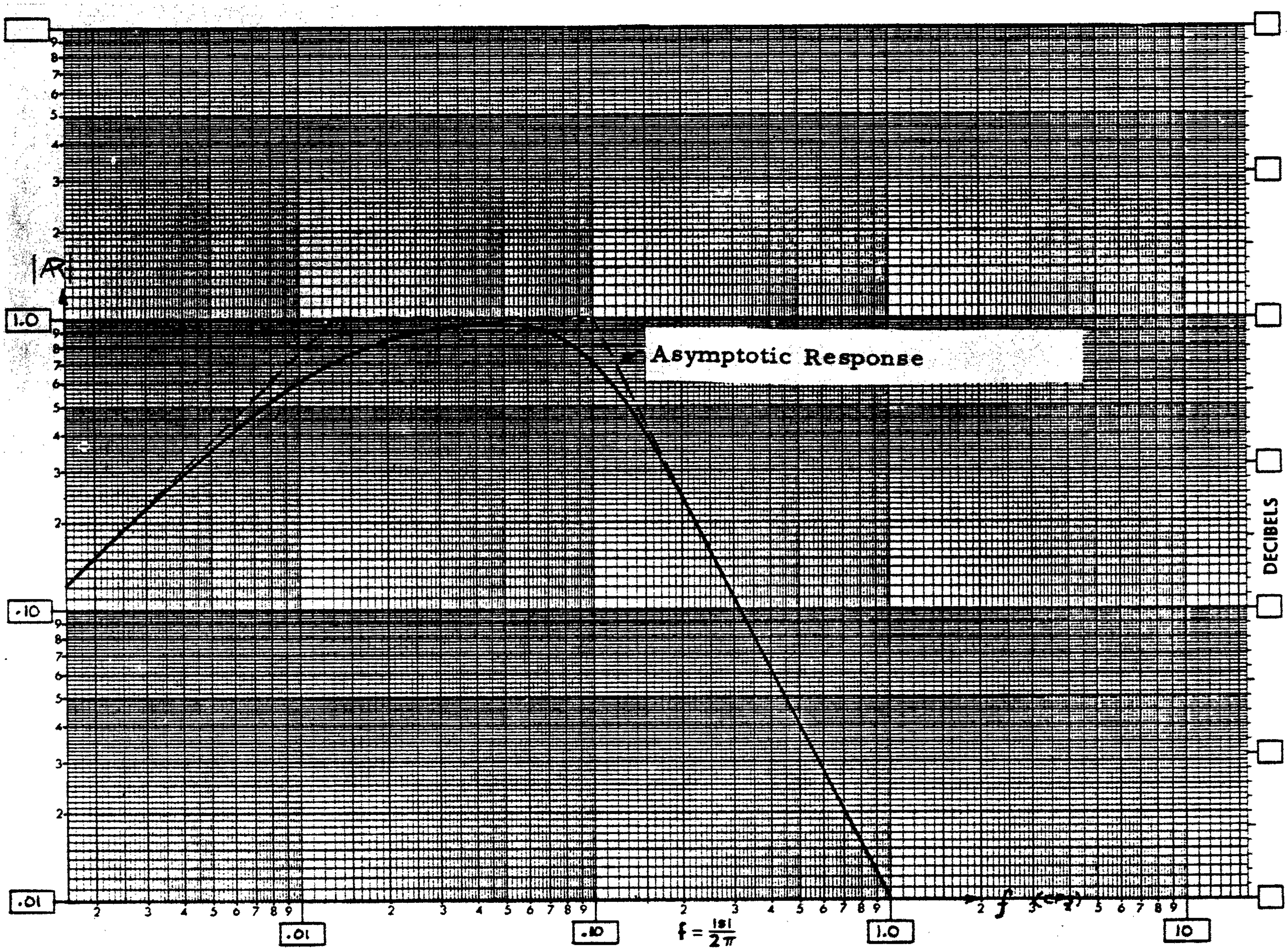

Figure 17 Random Noise Filter Amplitude-Ratio Versus Frequency

The voltages representing the motions of the input pointer and the CAL servo power arm were introduced into an analog computer which performed computing operations on the error signals. Continuous time histories of input-pointer motion, servo power arm position, and error (i. e., the vector differences between these quantities) were recorded for each test. A sample record is reproduced in figure 18. The duration of each test and the value of the time integral of the error squared were also recorded to permit computation of the root-mean-square (rms) error for each test. The rms error was used as the index of performance in comparing and evaluating test results although it was recognized that rms values give no information on the time history or peak values of the error.

Six male subjects were selected to perform the tests. Each subject was asked to track the pointer motions with the CAL servo power arm for several load conditions (viz. 0, 12.5, 25 and 50 lbs at the end of the power arm) for two minutes, with and without power boost. Each subject performed only once for each test condition. For the tests conducted without power boost, the output arm was rigidly fastened to the input arm and disconnected from the actuator strut. With power boost, all six subjects were capable of tracking the full two minutes at each load mentioned above with no difficulty. Without power boost, all subjects again tracked the full two minutes in the no-load case, only one tracked two minutes at 12.5 lbs, no one succeeded in tracking two minutes at 25 lbs, and no one succeeded in tracking 50 lbs for any duration. The raw data are given in table IX.

The test results are indicated in figure 19, where the mean of the rms errors for the six subjects is plotted as a function of load weight. Also shown on figure 19 is the spread between the maximum and minimum rms errors at each load condition.

The average rms error for all subjects at no load and without power boost was 1.59°; with additional load weight, the error increased slightly to a maximum of 2° with a load of 25 lbs. This 26% increase is not particularly significant from an accuracy standpoint; the small increment in error implies that the subjects were capable of tracking the particular input function with only a small degradation in accuracy. The significance here is that the fatigue limit was reached by all subjects at 25 lbs long before the desired two minute test duration had expired. In other words, if the subjects could track with a given load at all, they could track with an accuracy only slightly less than the accuracy obtained at no load. Note, however, that with power boost, the subjects not only were capable of manipulating any load within the test range with the same accuracy (about 1.6° rms error) as in the no-load case, but also that this accuracy is essentially the same as that obtained at no load with no power boost (this latter condition is essentially a measure of the best possible response to a truly random input for each subject using no auxiliary equipment).

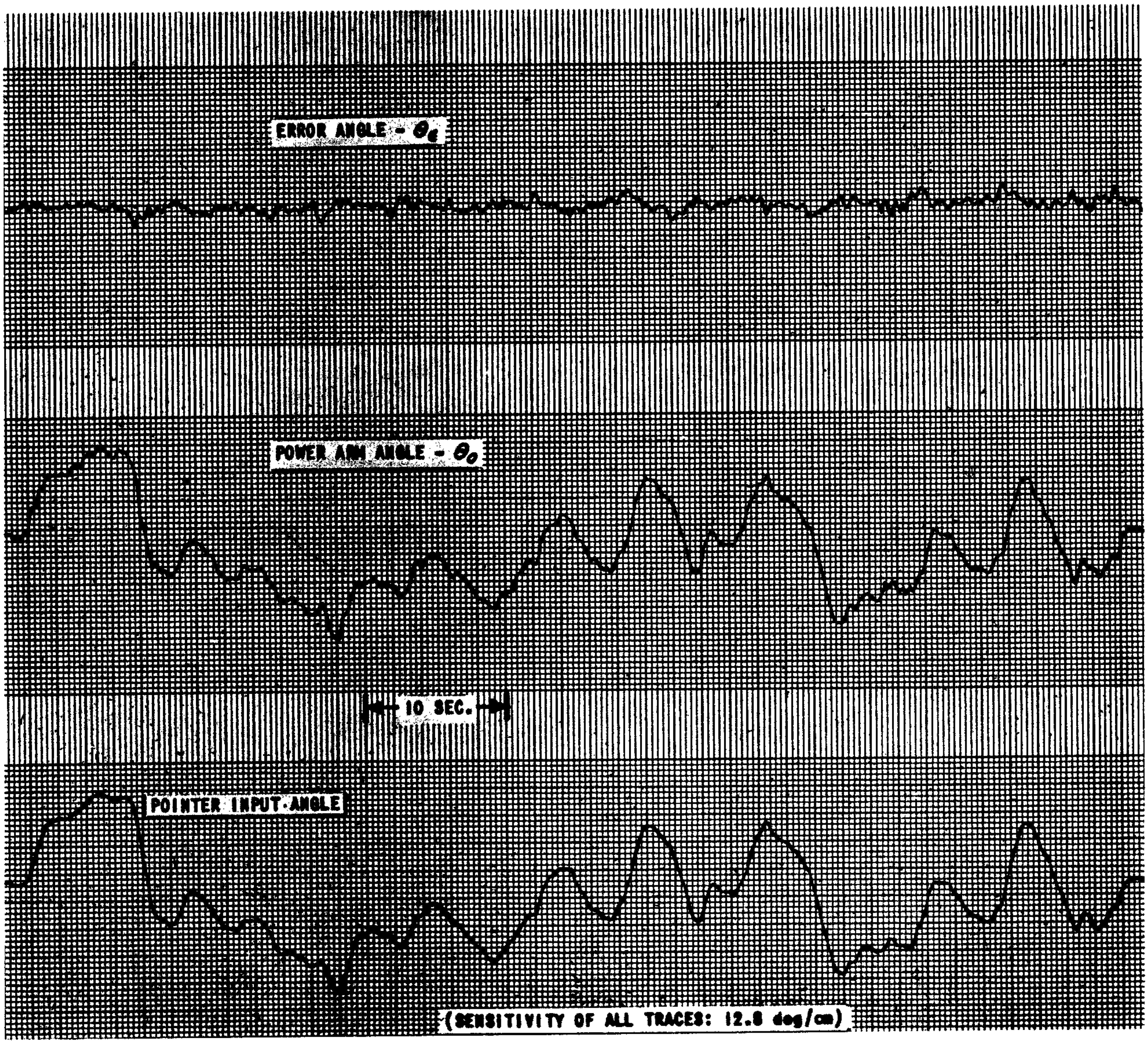

Figure 18 SAMPLE TRACKING TEST TIME HISTORY

TABLE IX

TRACKING TEST DATA

Subject	Load lbs.	Test Time* sec. (No Boost)	Error deg.-rms No Boost	Error deg.-rms Boost
1	0	120	1.60	1.65
	12.5	41	1.50	1.57
	25	24	2.36	1.60
	50	0	----	1.54
2	0	120	1.49	1.43
	12.5	66	1.62	1.49
	25	38	1.54	1.40
	50	0	----	1.52
3	0	120	1.82	2.02
	12.5	61	2.15	1.91
	25	0	----	1.89
	50	0	----	1.82
4	0	120	1.43	1.57
	12.5	72	2.07	1.62
	25	106	2.46	1.80
	50	0	----	1.78
5	0	120	1.60	1.60
	12.5	74	1.99	1.70
	25	40	1.90	1.75
	50	0	----	1.78
6	0	120	1.60	1.60
	12.5	120	1.75	1.68
	25	41	1.73	1.51
	50	0	----	1.49

* Test time with power boost = 2 min.

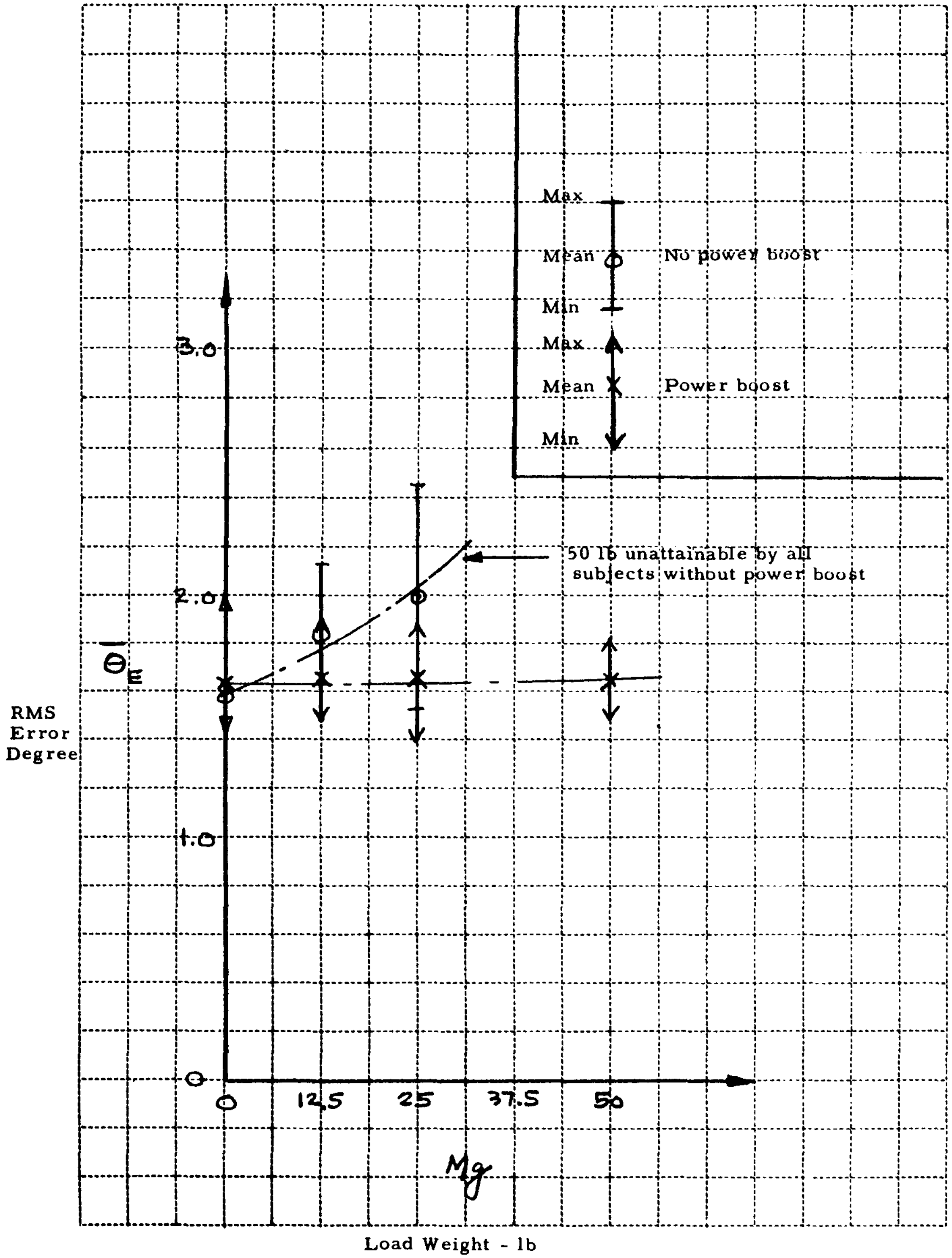

Figure 19 Mean Tracking Error Versus Load Weight

CONCLUDING REMARKS

The results of the exploratory investigations discussed herein do not permit firm and final conclusions to be drawn as to the technical feasibility of the Man Amplifier concept. As a general conclusion, it may be stated that investigations to date have not revealed the concept to be definitely unfeasible.

The studies of (1) the human body, (2) the kinematic functioning of the various joints, and (3) the complex motions which can be performed with the motion ranges and the many degrees of freedom of the limbs have led to the conclusion that it would be impractical to attempt to duplicate within a Man Amplifier all of the joints and motion ranges of the human body. It will be necessary, therefore, to select and incorporate in the Man Amplifier exoskeleton only those motion freedoms essential to the satisfactory operation of the device. It is further concluded that this selection must be made largely on the basis of experimental results because of the difficulty in perceiving and evaluating all of the consequences of a restricted motion capability.

For tasks requiring the Man Amplifier to lift and carry heavy loads, the capacity limit appears to be dictated more by considerations of stability than of structural strength. Static stability is a function of the geometrical situation, and hence is affected by the size and shape of the load and the ability of the Man Amplifier to get close to the load center-of-gravity. In general, it is doubtful that loads greater than about 1000 lbs could be lifted and carried by a single Man Amplifier. In view of this limited capacity, "amplified" men working independently could not replace the greater-capacity cargo-handling vehicles used by the Air Force today. However, two or more "amplified" men working as a team could have a combined capacity greatly exceeding the sum of their individual load limits.

Results of preliminary servo system studies indicate that the no-load velocity requirement and the maximum force output requirement of the Man Amplifier cannot both be satisfied by hydraulic servos employing a conventional servovalve for control without wasting a large amount of power. This wasted power would result in a low operating efficiency and an unacceptably large power output requirement for the prime mover. The delivered output power of the servos in performing a typical lifting task is estimated to be approximately 10 horsepower. However, provision for maximum load power to all servos simultaneously (100% duty cycle), based on the sum of the estimated maximum requirements for each servo in performing a typical lifting, supporting and transporting task, will require a power supply capable of delivering 25 to 30 horsepower or more, depending upon the number of servos employed. This value assumes that a power supply can be designed to provide modulated flow directly to each servo actuator as a function of the servoloop error and hence, operate at nearly 100% efficiency for all load force-velocity conditions.

Insofar as the structural and mechanical design aspects of the Man Amplifier are concerned, present indications are that the use of currently available, high-strength, aluminum alloys will be adequate to carry the loads imposed on the exoskeletal structure and that the overall weight of the device, excluding the weight of the operator, will be in the vicinity of 400 to 500 lbs. It is clear, however, that considerable ingenuity will be required in the design of the various elements and components comprising the exoskeleton to provide sufficient strength while at the same time satisfying the requirements for size adjustability, motion freedoms, safety, ease of donning and doffing, etc., without exceeding space limitations.

As a result of the work performed with the CAL elbow-joint servo in the experimental human-factors investigations, it is concluded that an elbow-joint servo having a frequency response of four cycles per second or greater is adequate for any reasonable tracking task. Results of tracking tests with the CAL elbow-joint servo show tracking error under power-boost conditions at all loads unchanged from that under the no-boost, no-load conditions; i.e., the servo did not measurably increase the tracking error above that of the unaided operator.

Multiple feedback-loop servos, including force-feel, must be incorporated into each joint servomechanism to achieve force and position accuracy and overall stability. Stability problems caused by servo load forces coupling back into the servoloop through mechanical and hydraulic compliance will almost certainly be encountered in the design of each Man Amplifier servo. These undesired compliances will necessitate refined design of the skeletal structure and hydraulic components. If gaseous working fluids are used, these problems could be even more severe. Fortunately, servo-stabilization techniques such as dynamic pressure feedback or transient-flow stabilizers are available to apply to this projected servo stability problem.

Though difficult problems are recognized in virtually every major area connected with developing the Man Amplifier, the tasks briefly outlined below are thought to be necessary to further assess the feasibility of the concept and contribute to its ultimate development.

1. In order to determine the minimal number of joints, their locations, and their motion ranges, for the Man Amplifier structure, a full-scale, wearable mockup of a nonamplifying, exoskeletal structure should be designed, fabricated and tested. This structure should be capable of adjustment for (a) wear by a number of subjects and (b) limitation of the motion range of each joint. Tests with human subjects would provide subjective and objective data concerning the effects of various degrees of movement restriction on performance of representative tasks. These data could be used to define the joint freedom requirements of a Man Amplifier type exoskeleton. At the same time, such a program would be very valuable in providing background and insight to mechanical design problems (and their solution) likely to be encountered in the design of a powered exoskeleton.

2. Studies should be performed to analyze the servo system requirements of the Man Amplifier with respect to sensors and sensor-servo interconnections required to implement a multi-degree-of-freedom response to human-limb motions. These studies should investigate what parameters need to be sensed, methods and devices used for sensing, the location and arrangement of sensors, and problems associated with the signal processing required to link the sensor outputs with the appropriate servos.

3. Finally, an analysis should be made of the servo system requirements of a Man Amplifier device with respect to servo system performance and stability. Servo systems for each joint should be postulated and analyzed to select the optimum form of servo-feedback loops. Attempts should be made to evolve, by varying the servo characteristics as required, servomechanisms that will allow stable operation of each controlled member, both singly, and in combination with adjacent members.

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APPENDIX A

ANALYSIS OF THE CAL ELBOW-JOINT SERVO

A mathematical analysis was performed to define the operational characteristics of the CAL elbow-joint amplifier (hereafter called CAL Servo). A summary of the force balance and kinematic restraint equations, and an indication of the method of generating these equations, is presented below.

In figure 20, a mechanical equivalent-circuit of the CAL Servo is shown that relates the servovalve oil "displacement"* with the valve body displacement, servopiston motion, and load movements. In this figure, oil compressibility and hydraulic line compliance are lumped and represented by the spring k_e. The elements C_T and k_T constitute a transient flow stabilizer (ref. 8) which was added to augment servoloop stability and is described later in this section. The lever arm is used to represent the various mechanical advantages involved in the servoloop. The spring k_ℓ represents the stiffness of the power arm, and M is the load mass concentrated at the outer end of the power arm. It was assumed that the power arm structural resonance is very lightly damped; hence, zero damping was used in the analysis. Actually, any structural damping tends to stabilize the servoloop.

The force-balance equations applicable to the system shown in figure 20 yield the following transfer function:

$$\frac{x_o}{\delta_e}(s) = AB\,\frac{MC_T s^3 + Mk_T s^2 + k_\ell C_T s + k_\ell k_T}{MC_T\left[1+\frac{k_\ell}{k_e}(B\ell)^2\right]s^3 + M\left[k_T + k_\ell(B\ell)^2 + \frac{k_T k_\ell}{k_e}(B\ell)^2\right]s^2 + C_T k_\ell s + k_\ell k_T} \qquad (7)$$

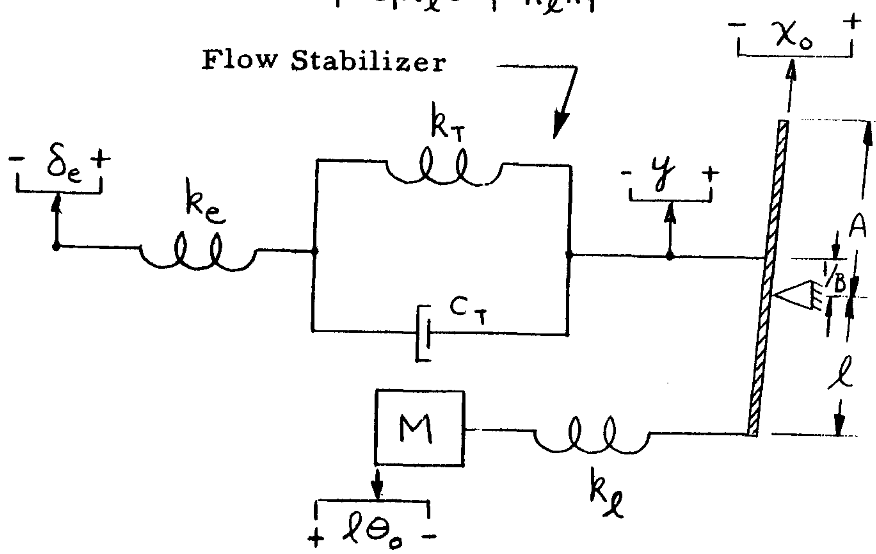

Figure 20 Mechanical System Equivalent Circuit for CAL Elbow-Joint Servo

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* Servovalve oil displacement is the time integral of the ratio of servovalve flow to actuator piston area. That is: $\delta_e = \int_0^t \left(\frac{Q_V}{a}\right) dt$

It is convenient to write Eq (7) in a different form to facilitate the analysis. Equation (8) shows the effect of oil and hydraulic-line compliance and servopiston leakage on one element of the CAL Servo forward loop.

$$\frac{\chi_o}{\delta_e}(s) = AB \frac{1 + \left(\frac{C_T}{k_T}\right)s + \left(\frac{M}{k_\ell}\right)s^2 + \left(\frac{MC_T}{k_\ell k_T}\right)s^3}{1 + \left(\frac{C_T}{k_T}\right)s + M\left[\frac{1}{k_\ell} + (B\ell)^2\left(\frac{1}{k_T} + \frac{1}{k_e}\right)\right]s^2 + \frac{MC_T}{k_\ell k_T}\left[1 + \frac{k_\ell}{k_e}(B\ell)^2\right]s^3} \tag{8}$$

Note that when compliances, k_e and k_T, possess infinite stiffness, a frequency-independent transfer function is obtained. In this case, no load reactions are transmitted into the servoloop and the closed-loop response is unconditionally stable for all values of load mass and power-arm stiffness. This is an ideal case, however, and one which is impossible to achieve in practice. Actually, the compliances of the oil and the mechanical members combine to allow deflections of the power arm away from the servo null position. Depending on the characteristics of the power-arm load, the internal compliance can be very detrimental to stable closed-loop operation of the servo. This problem is considered further in subsequent discussions.

The servoloop can be represented in block diagram form as shown in figure 21.

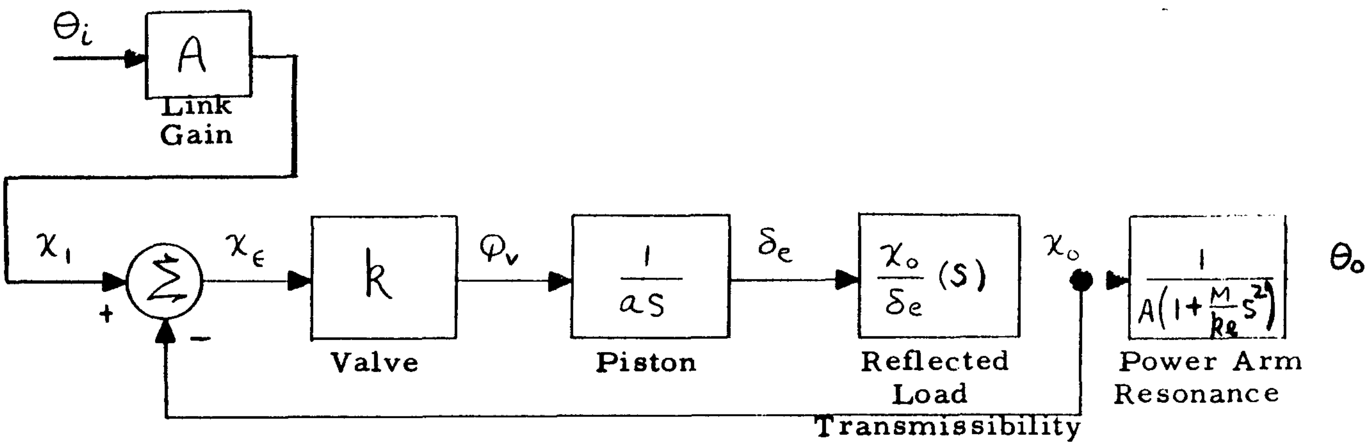

Figure 21 Servoloop Block Diagram

This figure indicates that the closed-loop transfer function, $\frac{\chi_o}{\chi_1}(s)$ is:

$$\frac{\chi_o}{\chi_1}(s) = \frac{\frac{k}{as}\ \frac{\chi_o}{\delta_e}(s)}{1 + \frac{k}{as}\ \frac{\chi_o}{\delta_e}(s)} \tag{9}$$

If the loop gain (velocity constant) is defined by $K_V \triangleq \frac{kAB}{a}$ and both numerator and denominator of Eq (9) are divided by this gain, Eq (10) results:

$$\frac{\chi_o}{\chi_I}(s) = \frac{1}{1 + \dfrac{AB\,s}{K_V \cdot \frac{\chi_o}{\delta_e}(s)}} \tag{10}$$

Substituting Eq (8) in Eq (10) yields the closed-loop transfer function.

$$\frac{\chi_o}{\chi_I}(s) = \frac{1 + \frac{C_T}{k_T}s + \frac{M}{k_\ell}s^2 + \frac{MC_T}{k_\ell k_T}s^3}{1 + \left(\frac{C_T}{k_T} + \frac{1}{K_V}\right)s + \left(\frac{M}{k_\ell} + \frac{C_T}{K_V k_\ell}\right)s^2 + \left[\frac{MC_T}{k_T k_\ell} + \frac{M}{K_V}\left\{\frac{1}{k_\ell} + (B\ell)^2\left(\frac{1}{k_\ell} + \frac{1}{k_e}\right)\right\}\right]s^3 + \frac{MC_T}{K_V k_T k_\ell}\left[1 + \frac{k_\ell}{k_e}(B\ell)^2\right]s^4} \tag{11}$$

Eq (11) is most useful for analyzing servo performance because it defines closed-loop stability, transient response, and frequency response in terms of the physical parameters of the servo. The denominator of the equation is the characteristic equation of the system and governs servo stability. Application of Routh's criteria (ref. 9) to the characteristic equation results in the following requirement for a stable servoloop:

$$K_V < \frac{\beta}{2\alpha} + \sqrt{\left(\frac{\beta}{2\alpha}\right)^2 + \frac{\gamma}{\alpha}}$$

where:

$$\alpha = \frac{C_T}{k_T}\left[\frac{M}{k_\ell}\left(\frac{1}{k_T} + \frac{1}{k_e}\right) + \frac{1}{k_e}\left(\frac{C_T}{k_T}\right)^2\right]$$

$$\beta = \left(\frac{C_T}{k_T}\right)^2\left(\frac{1}{k_T} - \frac{1}{k_e}\right) - M\left(\frac{1}{k_T} + \frac{1}{k_e}\right)\left[(B\ell)^2\left(\frac{1}{k_T} + \frac{1}{k_e}\right) + \frac{1}{k_\ell}\right]$$

$$\gamma = \frac{1}{k_T}\;\frac{C_T}{k_T}$$

It is apparent that both α and γ will always be positive numbers (or zero) since the servo parameters are all positive numbers. When α and γ are not zero, a stable loop gain can always be found, regardless of the sign of β Larger loop gain and better closed-loop performance are possible when β is positive. Non-zero values of α and γ are possible only when C_T/k_T is a finite positive number.

If C_T/k_T is zero, the servo will not be stable for a finite load inertia condition, at any loop gain. This situation was encountered with the CAL Servo in the "debugging" period.

An analog computer study of a transient flow stabilizer (figure 22) applied to the CAL Servo indicated that the flow stabilizer provided stable servo operation only for relatively small loop gains. For example, stable operation could not be achieved without reducing the loop gain, K_V , to 25 sec^{-1}. Even at such a small loop gain, the damping is small, resulting in a resonant response.

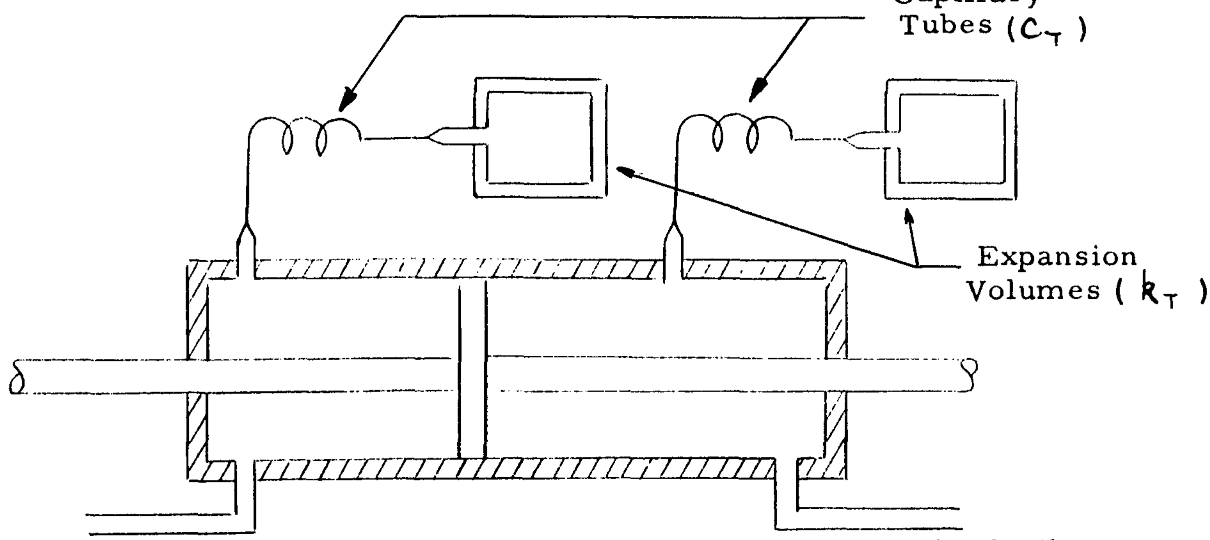

Figure 22 Transient-Flow Stabilizer

The stabilizer time constant was selected to maximize the internal servoloop energy loss at the natural frequency of the servo while keeping the servoloop stiffness as large as possible. A pair of capillary tube/oil chamber stabilizers was constructed using the analog computer results as a basis for design criteria. These units were of some help, but only for small values of loop gain. Because the servo velocity-response was very poor at these low loop gains, it was decided to seek a more satisfactory solution to the stability problem.

A convenient way to obtain internal damping in a hydraulic servoloop is to provide a leakage path across the servopiston. By so doing, energy is extracted from the servoloop in proportion to the flow through the by-pass. To provide this leakage path in the CAL Servo, a by-pass valve was installed directly across the servopiston. Under the most detrimental load impedance condition (viz. a large inertia load), the by-pass valve was opened until unconditionally stable operation resulted.

The closed-loop system transform for the case of a "leaky" servopiston can be represented by Eqs (7) and (9) with $k_T = 0$ in Eq (7). This transform is given by:

$$\frac{x_o}{x_i}(s) = \frac{1 + \left(\frac{M}{k_e}\right)s^2}{1 + \left(\frac{1}{K_v}\right)s + M\left[\frac{1}{k_\ell} + \frac{(B\ell)^2}{K_v C_T}\right]s^2 + \frac{M}{K_v k_\ell}\left[1 + \frac{k_\ell}{k_e}(B\ell)^2\right]s^3} \tag{12}$$

Application of Routh's stability criteria to the denominator of Eq (12) results in the following condition for stable operation.

$$K_v < \frac{k_e}{C_T}$$

Recalling that C_T is the damping due to by-pass valve opening, and that this damping is inversely proportional to the valve opening, it is apparent that large by-pass flows produce small values for the damping constant with large energy loss. This explains why the damping term should appear in the denominator of the above inequality and also why small values of damping obtained by this method result in greater servoloop stability. Note also that allowable loop gain increases directly with hydraulic and mechanical stiffness. It is apparent that a given desired value of loop gain can always be achieved, for any internal compliance, by adjusting the magnitude of the damping term (i. e., opening the by-pass valve). The above described method of stabilization has the disadvantages of wasted power, reduced maximum servo force output, and reduced maximum actuator velocity at any given load, all of which result from the required leakage flow. For the CAL Servo, the advantages were judged to outweigh these disadvantages. A more commonly accepted method used to eliminate servo instability caused by reflected load impedance is "dynamic pressure feedback". This method avoids undue power loss and wasted flow but could not be readily applied to the all-mechanical CAL Servo.

APPENDIX B

SINUSOIDAL FREQUENCY RESPONSE CHARACTERISTICS

One of the commonly accepted criteria for assessing servomechanism performance is the sinusoidal frequency response of the system. This criterion was used in a comparison of the theoretical and measured performance of the CAL Servo.

1. Numerical Parameter Values

The parameters of the servoloop govern the behavior of the servo in responding to arbitrary forcing functions. For the CAL Servo, many of the system parameter values were directly measured; others were estimated when their direct measurement was difficult or impractical to obtain. The nominal values used in the sinusoidal frequency response analysis are listed in table X.

2: Calculated Response

The sinusoidal frequency response of the CAL Servo can be determined from an evaluation of the following equation:

$$\frac{\theta_o}{\theta_i}(j\omega) = \lim_{s \to j\omega} \left\{ \frac{\chi_o}{\chi_1}(s) \cdot \frac{\theta_o}{\chi_o}(s) \cdot \frac{\chi_o}{\theta_i}(s) \right\}$$

$$\frac{\theta_o}{\theta_i}(j\omega) = \frac{1}{1 - M\left[\frac{1}{k_\ell} + \frac{(B\ell)^2}{K_v C_T}\right]\omega^2 + j\,\omega\left[\frac{1}{K_v} - \frac{M}{K_v k_\ell}\left\{1 + \frac{k_\ell}{k_e}(B\ell)^2\right\}\omega^2\right]} \qquad (13)$$

The magnitude and phase angle of the steady-state power-arm response to sinusoidal motions of the input arm, are predicted by Eq (14). Defining $M(\omega)$ and $N(\omega)$ as even functions of angular frequency, as below,

$$\left.\begin{aligned} M(\omega) &= 1 - M\left\{\frac{1}{k_\ell} + \frac{(B\ell)^2}{K_v C_T}\right\}\omega^2 \\ N(\omega) &= \frac{1}{K_v}\left[1 - \frac{M}{k_\ell}\left\{1 + \frac{k_\ell}{k_e}(B\ell)^2\right\}\omega^2\right] \end{aligned}\right. \qquad (14)$$

and substituting these expressions into Eq (13), the following equation is obtained:

$$\left.\begin{aligned} \left|\frac{\theta_o}{\theta_i}(\omega)\right| &= \frac{1}{\sqrt{M^2(\omega) + \omega^2 N^2(\omega)}} \\ \varphi(\omega) &= \arctan\left\{\frac{-\omega N(\omega)}{M(\omega)}\right\} \end{aligned}\right\} \qquad (15)$$

TABLE X

NOMINAL VALUES FOR SERVO PARAMETERS

P_s	=	3000 psi (measured)	
k	=	$445 \frac{in^3/sec}{in}$	(Manufacturers data)
A	=	0.970 in/rad (measured)	
B	=	0.286 rad/in at $\Theta_o = 0$ 0.374 rad/in at $\Theta_o = 60°$	(computed; a nonlinear function of power arm amplitude)
a	=	$0.59\ in^2$ (measured)	
ℓ	=	18.5 in. (measured)	
M	=	$0.0078\ lb\text{-}sec^2/in$ min $0.137\ lb\text{-}sec^2/in$ max	(measured power arm load mass plus calculated arm mass)
k_ℓ	=	820 lb/in	(computed based on assumed material properties and measured dimensions)
k_e	=	20,000 lb/in	(computed based on assumed oil bulk modulus, calculated strut and housing stiffness, calculated oil volume, and assumed frame stiffness)
C_T	=	30 lb-sec/in	(an estimate based on measured velocity of power arm in response to a specified force)
K_V	=	$210\ sec^{-1}$; $\Theta_o = 0$ $273\ sec^{-1}$; $\Theta_o = 60°$	(computed from above numbers)

Equations (15) represent the system response separated into the desired amplitude ratio and phase angle as functions of frequency.

The frequency response of the CAL Servo was calculated using Eqs (14) and (15) and the numerical values listed in table X. Calculations were made for both no-load and 50 lb load conditions. With no load on the power arm of the servo, M is zero; $M(\omega) = 1$; and $N(\omega) = \frac{1}{K_V}$. The response of the servo is that of a first-order system having a corner frequency at $K_V/2\pi$ cycles per second (cps). For $K_V = 210\ sec^{-1}$, the corner frequency is 33.4 cps. This shows that, with no load, the servo performance is excellent to frequencies well beyond human input capabilities. The theoretical no-load amplitude response is plotted in figure 23. The theoretical phase angle response appears in figure 24.

With a 50 lb weight attached to the end of the power arm, the servo response is modified such that a resonant peak occurs at 6.5 cps. This resonant peak is caused by the load inertia "reflecting" back into the servo-loop through the compliant elements in the system. As indicated in the stability analysis performed in Appendix A, the servo would be unstable without the servopiston by-pass valve opened slightly. The full-load amplitude ratio and phase angle frequency response plots are also shown in figures 23 and 24.

3. Measured Response

To obtain an indication of the accuracy of the theoretical results just presented, the amplitude-ratio frequency response of the CAL Servo was measured for the no-load and 50 lb load conditions. No attempt was made to measure phase angle, because it can be computed, if desired, from the amplitude-ratio results (ref. 10).

A photograph of the test setup is shown in figure 25. The CAL Servo was mounted on the bed of a milling machine to minimize any tendency of the servo to vibrate the support. As indicated in the photo, an electric motor was used to impart a sinusoidal motion to the input arm of the servo. The rotary motion of the motor shaft was converted to linear motion of the servo input arm by means of an eccentric drive wheel on the motor shaft. The frequency of the input motion was adjusted, up to the maximum available value of 30 cps, by varying the motor speed.

With no load on the servo, the measured frequency-response amplitude ratio was "flat" to 30 cps. With a 50 lb load, the resonant peak occurred at 8 cps as compared with the previously noted value of 6.5 cps computed from the theoretical analysis. Curves showing the measured frequency response characteristics are plotted in figure 26. A comparison of these results with the computed results presented in figure 23 indicates that the general form of the response was predicted reasonably well.

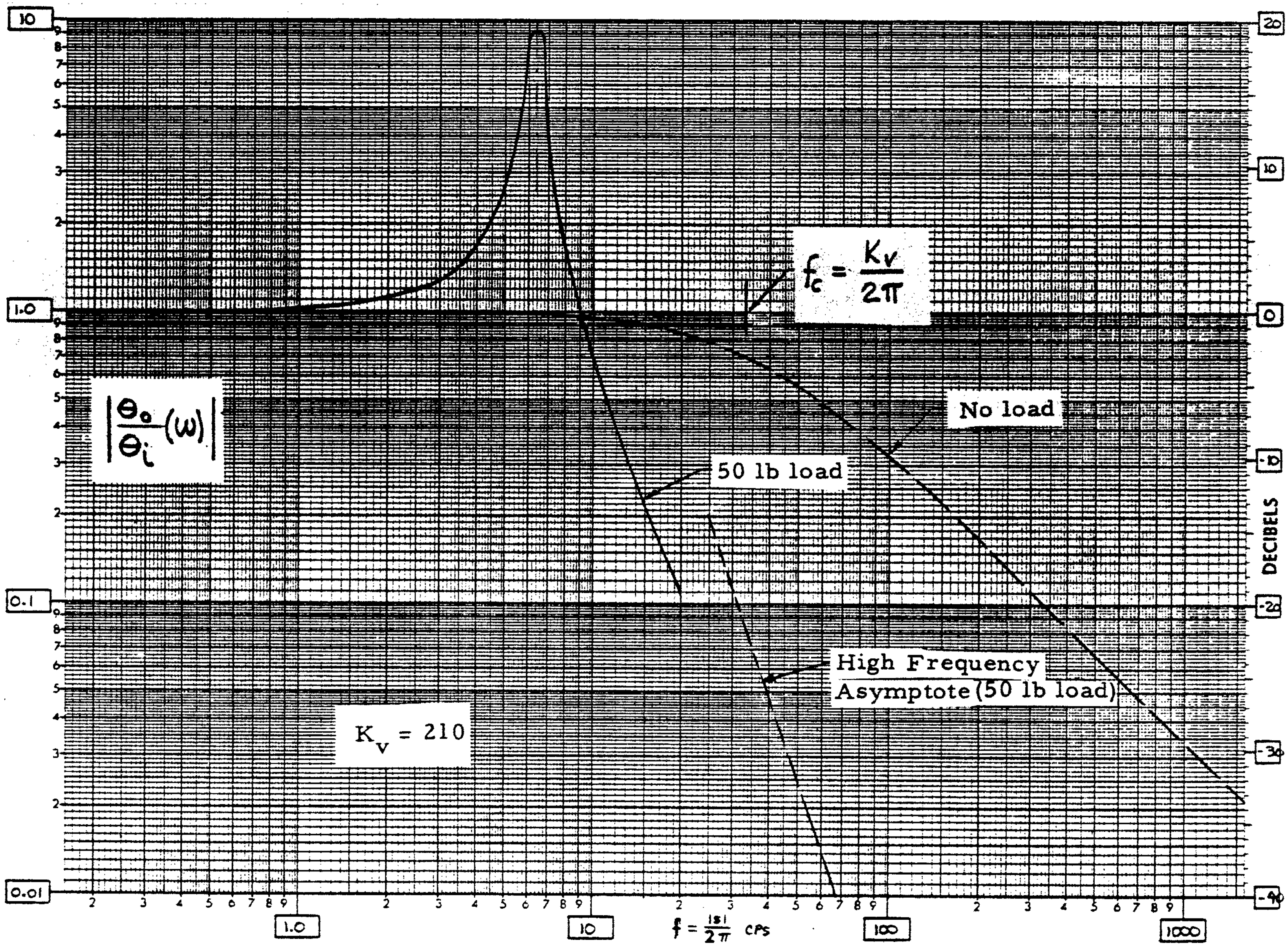

Figure 23 Theoretical Amplitude-Ratio Sinusoidal Frequency Response

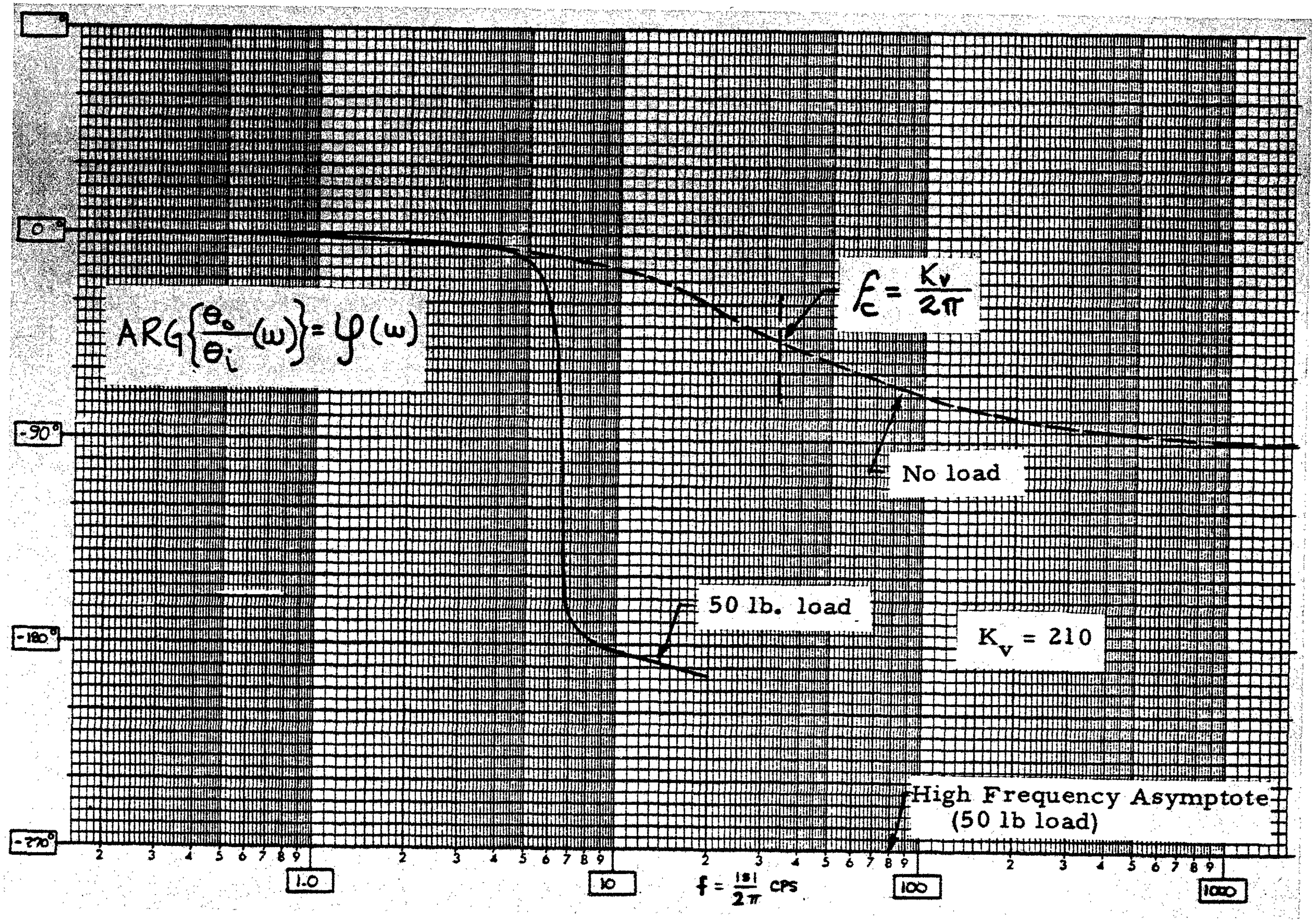

Figure 24 Theoretical Sinusoidal Phase Response

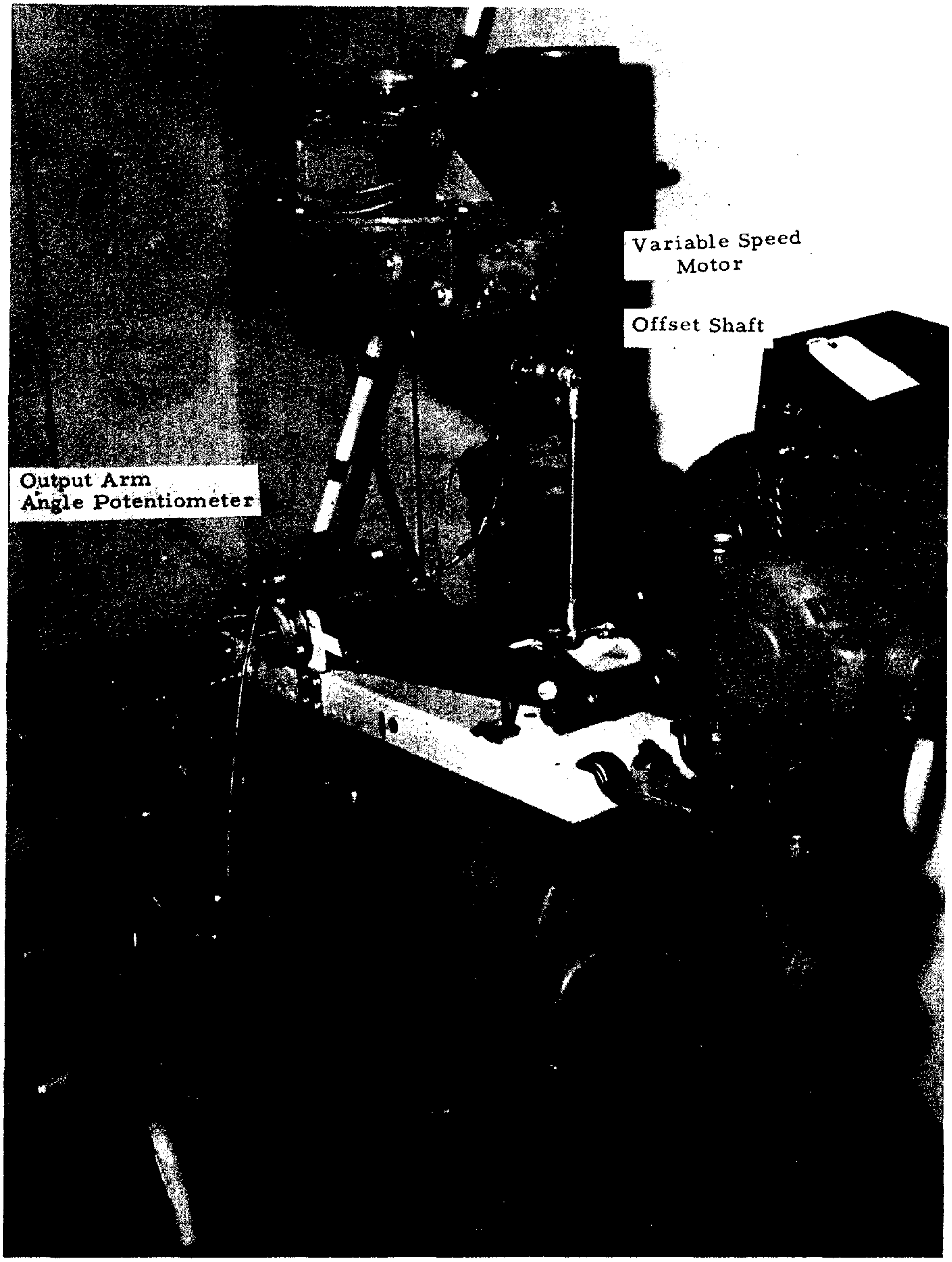

Figure 25 SINUSOIDAL FREQUENCY RESPONSE MEASUREMENT TEST SETUP

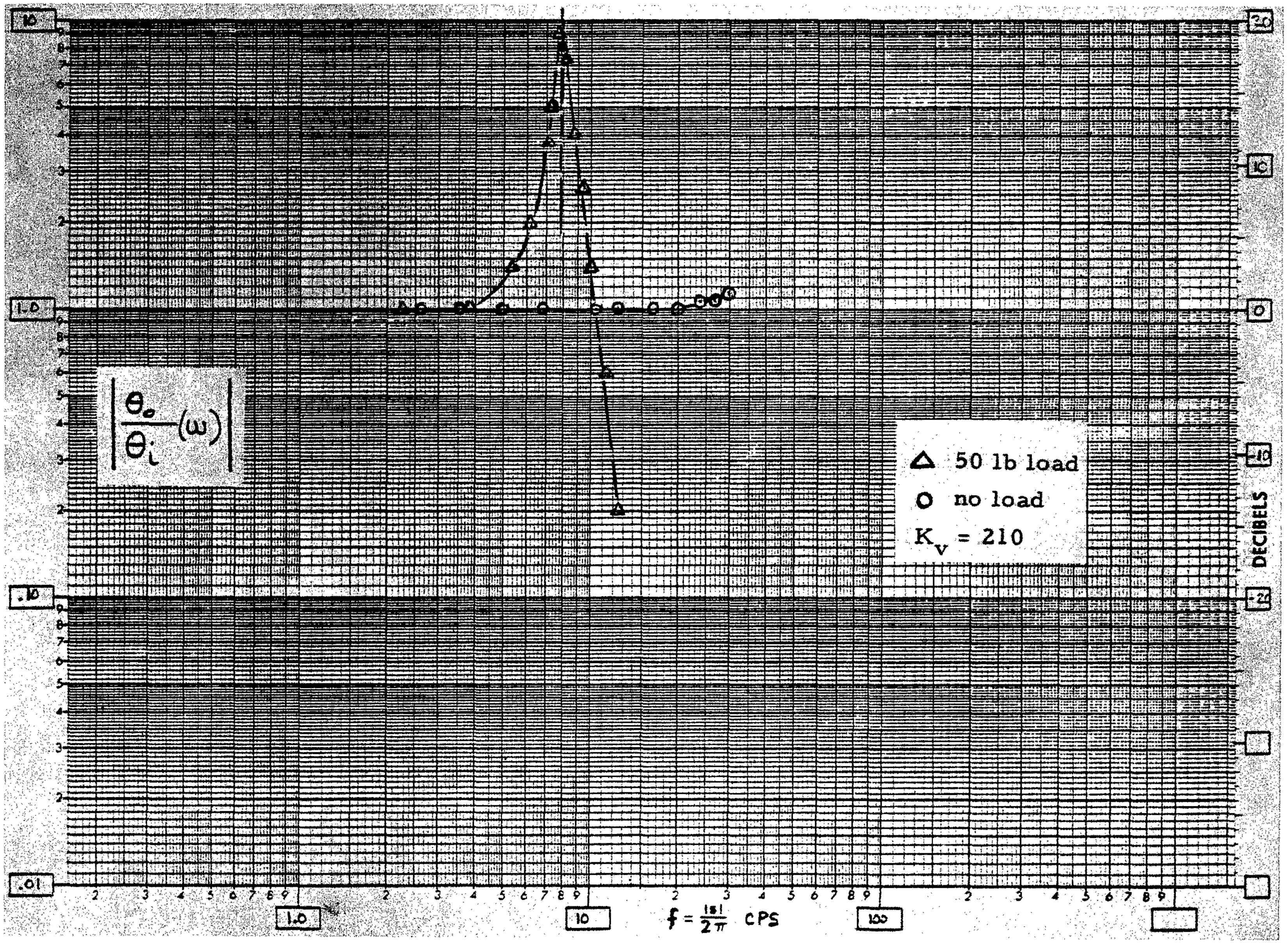

Figure 26 Measured Amplitude-Ratio Sinusoidal Frequency Response

APPENDIX C

EXPLANATION OF TERMS

Symbol	Description	Nominal Units
M	load mass	lb-sec^2/in.
C_T	flow stabilizer or by-pass valve damping rate	lb-sec/in.
k_e	internal servoloop spring rate	lb/in.
k_ℓ	power arm stiffness	lb/in.
k_T	flow stabilizer stiffness	lb/in.
ℓ	length of power arm	in.
a	servopiston area	in.^{2}
k	servovalve no-load flow constant	in.^{3}/sec/in.
B	power arm angular displacement per unit strut motion	rad/in.
A	servovalve spool displacement per unit error angle	in/rad
K_V	velocity error coefficient (loop gain)	sec^{-1}
α	Routh criteria stability parameters	in.-sec^3/lb
β	Routh criteria stability parameters	in.-sec^2/lb
γ	Routh criteria stability parameters	in.-sec/lb
AR	amplitude ratio	-
N	gear ratio ($N > 1$ assumed)	-
σ	servomotor volumetric efficiency	-
D_M	servomotor displacement per revolution	in.^{3}/rev
Q_V	servovalve oil flow	in.^{3}/sec
Q_S	power supply oil flow	in.^{3}/sec
P_L	load pressure	psi
P_S	supply pressure	psi

Symbol	Description	Nominal Units
x_I	valve spool motion due to input arm displacement with fixed output arm displacement	in.
x_o	valve spool motion due to output arm displacement with fixed input arm displacement	in.
x_ϵ	net value spool displacement; error signal	in.
δ_e	equivalent oil displacement measured at the servovalve	in.
y	servopiston strut displacement	in.
θ_i	input arm angular displacement	rad
θ_o	output arm angular displacement	rad
$\bar{\theta}_\epsilon$	rms angular displacement error	rad
θ_L	load angular position	rad
θ_M	servomotor angular position	rad
T_L	load torque	in. -lb
T_M	servomotor torque	in. -lb
$HP_{L,S}$	horsepower (L = load, S = supply)	hp
ω	angular frequency	rad/ sec
f	circular frequency ($f = \frac{\omega}{2\pi}$)	cps
S	Laplace transform variable	rad/ sec
j	imaginary operator ($j = \sqrt{-1}$)	-
$M(\omega)$	in-phase frequency response component	-
$\omega N(\omega)$	quadrature frequency response component	-
$\varphi(\omega)$	phase angle of frequency response vector	deg
g	earth gravity acceleration ($g = 386.4$ in. / sec^2)	in. / sec^2
η	efficiency	

Note: Dots over symbols signify time derivatives

Aerospace Medical Division, 6570th Aerospace Medical Research Laboratories, Wright-Patterson AFB, Ohio
Rpt. No. AMRL-TDR-62-89. EXPLORATORY INVESTIGATION OF THE MAN AMPLIFIER CONCEPT. Final report, Aug 61, vi + 68 pp. incl. illus., tables, 10 refs. Unclassified report

Preliminary investigations were conducted to ascertain some of major problems requiring more research before feasibility of Man Amplifier concept can be evaluated. Study areas included possible Air Force applications as basis for selecting maximum load-carrying capability, human factors from standpoints of body kinematics and physical anthropology, structures and mechanical design, and servosystem and power requirements. Dynamic response characteristics of elbow-joint amplifier, determined (over)

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1. Human Engineering
2. Man-Machine Systems
3. Man Amplifier
4. Anthropometry
5. Kinematics

I. AFSC Project 7184; Task 718406
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theoretically and experimentally, were compared. Comparison of position tracking tests performed with and without power boost provided by elbow-joint servo indicated that power boost did not increase tracking error above that exhibited by unaided operator. It was concluded: (1) duplication, in Man Amplifier, of all human joint motion capability is impractical; (2) experimentation is necessary to determine essential joints, motion ranges, and dynamic responses; (3) inability to counter overturning moments will, in many instances, limit load-handling capability; (4) conventional valve-controlled hydraulic servos are unsuitable for Man Amplifier; (5) particularly difficult problems will be encountered in general areas of mechanical design, sensors, and servomechanisms.

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